PUMPING MACHINERY

A TREATISE ON THE

HISTORY, DESIGN, CONSTRUCTION AND OPERATION OF VARIOUS FORMS OF PUMPS

BY

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PREFACE

SEVEN YEARS ago the course of lectures which forms the basis of this book was given as a required course to the mechanical engineering students at the University of Missouri. As a number of civil engineering students elected this, it occurred to the author that there was a demand for a book to cover the field as he had planned the course.

The general plan of the work is to give a brief historical outline of the development of pumping machinery which is closely connected with the history of the steam engine; then to examine and describe the action of a number of common forms of pumps, following this by the methods of design of pumping apparatus.

The book is intended for the use of those who have studied hydraulics, mechanics, and the strength of materials. It is intended to develop certain general principles of mechanics which are applicable to pumping machinery as well as to train the student in application of certain of the theoretical portions of the studies of an engineering course.

The first two chapters are devoted to the development of pumping machinery, because the author feels the need, of all engineers, of a more intimate knowledge of the great works of the early members of the profession. The pump is so closely allied with the steam engine in early times that this history traces the early forms of that machine. The historical works mentioned in the list of text-books in the bibliography as well as the historical articles given in the same catalogue have been indispensable in the preparation of these chapters.

The descriptive chapters on modern pumps have been prepared from the catalogues and bulletins of manufacturers and from current technical literature. The figures have been redrawn from these sources to fit them for this work. The half-tones have been made from photographs or cuts of publications kindly furnished by the various manufacturers. The

thanks of the author to these manufacturers are here publicly expressed. The various articles in the technical press which have been of assistance in the preparation of the lectures and this work are listed as bibliography. The author is indebted to the writers of these articles and texts and to their publishers for the aid he has received. These chapters have been incorporated into the book with certain tables from the catalogues, so that the student may have in his library a reference book which will not only give him the peculiarities of the different forms of special pumps, but the dimensions or other data may be found so that he may know what size of pump may be had for a certain purpose.

In the preparation of the theoretical part of the work of design the author has used the methods employed by him in courses in steam-engine design and hydraulic motors, adding thereto methods developed from "Die Pumpen," by Hartmann-Knoke and Berg, and "Die Zentrifugal Pumpen," by Fritz Neumann. These two books have been of great assistance in the preparation of the work. The theory relating to air-lift pumps differs from that given by Harris in his articles in the Proceedings of the American Society of Civil Engineers, which were originally considered.

The author especially wishes to express his thanks in this place, to his wife, Mary E. Lewis Greene, for her aid in the preparation of the manuscript and in proof-reading, as well as for her advice during the progress of the work.

Grateful acknowledgments are made to the following: International Steam Pump Company and to Mr. William Schwanhauser, Henry R. Worthington Company, the Allis-Chalmers Company, and Mr. Wiii Sando, the Snow Pump Works, The Holly Pump Works, and Mr. Decrow, The American Steam Pump Company, the Alberger Company, the Buffalo Pump Company, the Prescott Pump Company, Deane Brothers Pump Company, the National Board of Fire Underwriters, Mr. Francis Head, Prof. Lewis F. Moody, Mr. F. C. Dunlap, Mr. Lester French, and Mr. A. F. Rolf for their aid in furnishing data, illustrations and other materials.

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CHAPTER I

HISTORICAL DEVELOPMENT

THE pump, considered as any apparatus used to raise water, is one of the oldest machines known to man. Long before the Christian Era, when man emerged from the age of the hunter into that of the shepherd, he found it necessary to raise water from wells for his flocks in places where there were no pure streams. Indeed, even before this age there must have been devices to raise water from low levels to supply the personal needs of the hunter and those dependent on him.

Following the age of the shepherd came that of the farmer, when the demand for an apparatus to raise water was greater than before. The shepherd no longer wandered over the country in search of pasture, but now he cared for a definite tract of land to furnish the food supply for his flocks, his herds, and his family. Unfortunately it became necessary for him to use lands on which there was not a sufficient natural supply of water for irrigation, and he was compelled to lift water from low streams to these fields, so as to increase the yield from his land.

The next age was that of townsman and manufacturer. For protection, mutual aid, and comfort, man began to live in towns and cities. This necessitated supplying water for the common use by gravity from some higher source to fountains, by means of hand pumps from wells, or by utilizing natural springs. These town fountains, pumps, or springs are still prominent objects in the cities of the Old World, and also

in our own colonial towns. As the town developed the supply was carried into some of the houses. Finally, as the burgher became a manufacturer and was compelled to dig into the earth for his raw materials, it became necessary to clear his excavations of the surface and subsurface waters which filled them.

Modern civilization has demanded more and more as pumping machinery has been perfected, until to-day running water in unlimited supply is found not only in the houses of cities, large and small, but even the isolated farm often has its own water works, giving ample water to each room of its house, barn, and dairy. The latest demand is the supply, at a moment's notice, of large quantities of water, under great pressure, to the congested districts of trade in our large cities for fire protection.

Thus from earliest times may be traced a demand for some means of raising water for man; for his herds and lands; for the purpose of clearing his mines, and finally for his own personal convenience as well as the protection of his property. The pumping machine has developed from a very crude origin, it is true, yet its earliest types were so effective that they may be found in use to-day although in modified form.

The present work will not consider the sources of water supply, the gauging of the flow, the distribution of water, its analysis or purification, or any of the problems of hydraulics save those which are concerned with the raising of water from one level to another.

Two of the earliest forms of pumps are the **Shadoof** and the **Noria,** the former being common in Egypt, while the latter is found in China and along the Euphrates as well as on the Nile. The first of these is the ordinary well sweep seen on many old farms in this country. A leather, earthen, or woven bucket is attached to an arm by means of ropes or tree branches and ropes. This arm is tied to a crossbeam supported in crotches of tree trunks planted in the ground at the edge of some river or well. The arm supporting the bucket is counterweighted by a stone or a mud ball, so that there will be practically no

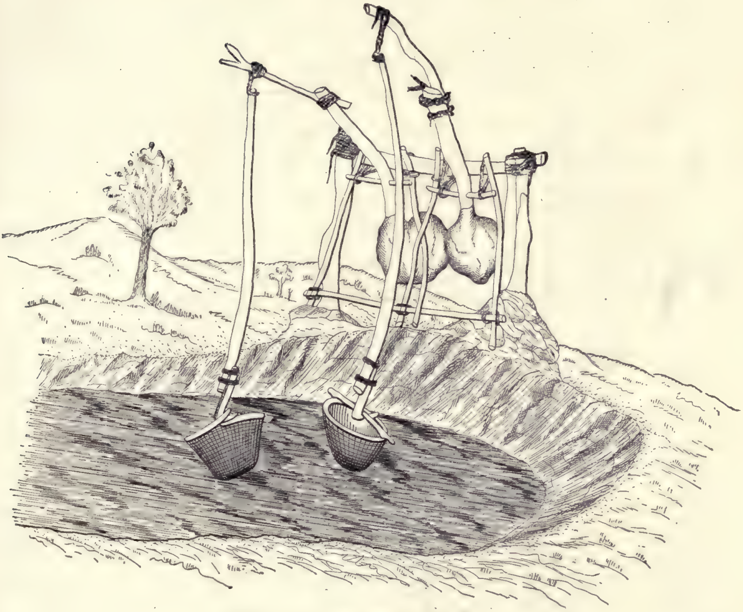


FIG. 1.—Shadoof or Noria.

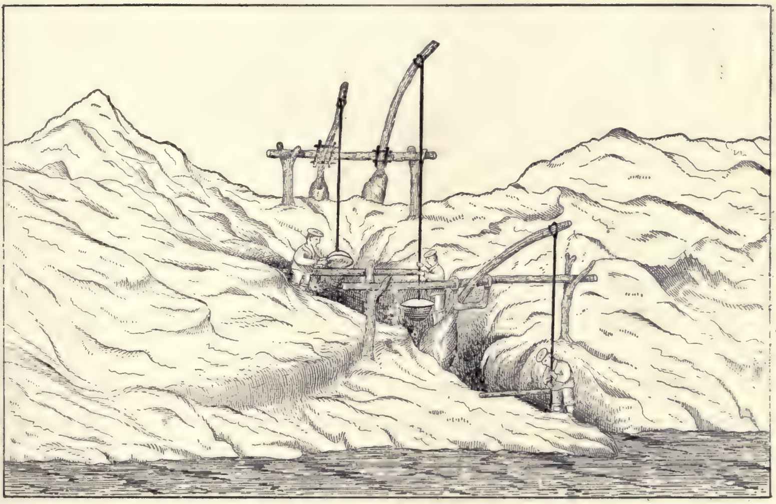


FIG. 2.—Shadoofs in Series.

weight to lift. A man then pulls on the bucket support, putting the bucket beneath the water, and then allows the counterweight to lift it to the proper level, where he empties the water into the canal or basin. From the canal it flows to the land which is to be irrigated. At times a series of these shadoofs is placed in line (Fig. 2), each shadoof raising the water a portion of the total lift which would be too great for any one machine. The shadoof is probably the oldest apparatus for raising water, although the simple bucket attached to a rope (Fig. 3) must have been used in early times. Wilson states that in India one or two men operate a *Paecottah* (the southern Indian name for the Egyptian shadoof), and lift from 1000 to 3000 cubic feet in from six to eight hours. The lift varies from 5 to 12 feet. This apparatus is known also as the *Swape* and in upper India it is called the *Lât.* Buckley states that the experiments of Wilson showed that two men could lift 57,600 cubic feet through a height of one foot in ten hours, while one man could lift 33,000 cubic feet.

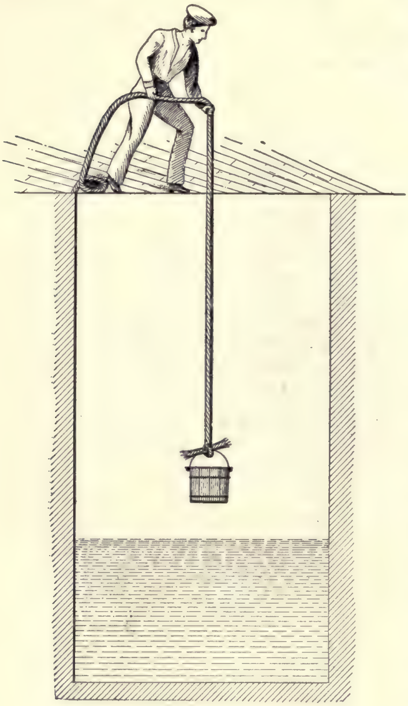


FIG. 3.—Bucket.

The Noria (Fig. 4) is a machine by which the water of a stream is raised in buckets attached to a wheel, the wheel being moved by the stream or in some cases by animal power. The Chinese claim to have used these as early as one thousand years before the Christian Era. One of them is described as consisting of eighteen or twenty arms with paddles, to the periph-

ery of which is attached a number of buckets. At the lowest limit of the motion of the wheel these buckets dip below the water and are filled. The motion of the wheel thus raises the water to a higher level. In some wheels the impact of the stream on the buckets is sufficient to drive the wheel, while in other cases the wheel is provided with additional vanes to drive it. A simple Chinese wheel (Fig. 4) is formed of bamboo. The spokes of this wheel are attached to a central shaft and cross each other at two-thirds the distance to the periphery,

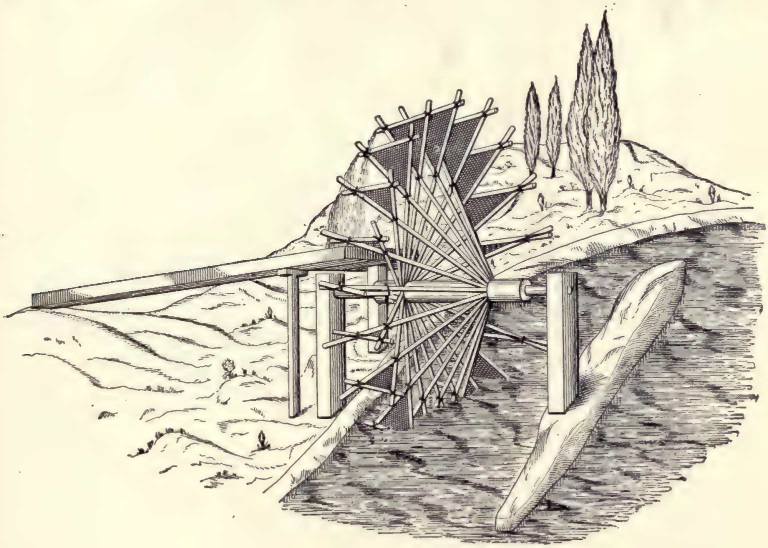


FIG. 4.—Noria.

where they are firmly lashed together, while at the end they are fastened to end pieces. These act as the buckets, the natural joint in the bamboo forming the bottom of the bucket. The triangle formed by the spokes and the end piece is filled with bamboo basket work, thus forming a paddle to drive the noria. The end tubes are cut and fastened at such an angle that when the vane is in a horizontal plane the end tube is inclined upward at an angle of twenty degrees so that the buckets will not discharge their contents until they pass the level of the axle. The buckets

finally discharge into a trough which conducts the water to a canal or reservoir from which it can be used. The wheels are from 20 to 40 feet in diameter. In the case of a wheel 20 feet in diameter, containing twenty buckets 2 inches in diameter and 4 feet long, 70,000 gallons were raised in twenty-four hours when the wheel made four revolutions per minute.

These wheels were used throughout the East in Asia and Egypt. Colonel Chesney of the British army reports several of them in operation along the Euphrates as motors and pumps. Some of these were arranged so that their axles could be raised by means of stones in order that their depths of immersion

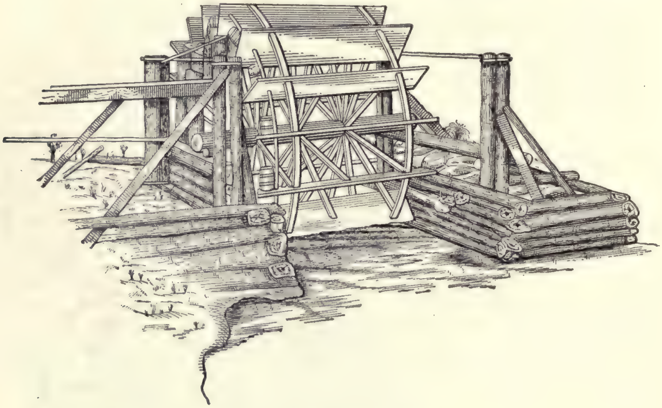


FIG. 5.—Modern Noria.

would be the same at different stages of the river. Along the river aqueducts are seen which carried the water away, and walls or dams are found which served to raise the water at the season of small flow and also served to direct the flow to the openings in the walls where the norias were placed. At the center of the river openings were left through which boats could pass.

An American wheel of modern application (Fig. 5) shows that the idea of centuries ago is still of value as the basis of an irrigating apparatus; as arranged here the full diameter is almost available for the lift. Such a machine is simple and very effective.

The name "noria" is also applied to a development of the older ancient machine, in which the buckets are attached to a chain or rope which, with its series of buckets or pitchers, is placed over the wheel and extends down to the water at considerable distance below. In this case, however, the wheel is driven by animal power. This form is known as the **Persian Wheel** (Fig. 6), while according to Buckley, the term *Sakias* is used for it in certain other places. To operate the Persian wheel a buffalo or camel is driven around a vertical axis, to an arm of which it is yoked. The axle contains a cog wheel (Figs. 6, 7, and 8) of crude form which engages with a second cog wheel mounted on a horizontal axis on which is placed the

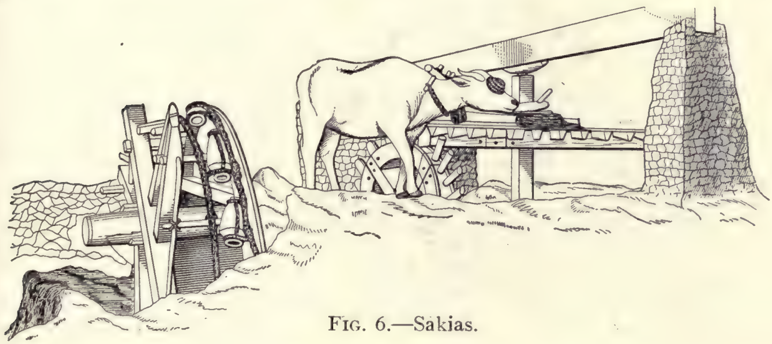


FIG. 6.—Sakias.

band or chain wheel for the support of the bucket chain. The water is discharged from the buckets into a trough and flows to whatever fields are to be irrigated. It is stated that two bullocks will lift 2000 cubic feet of water per day and the heights of the lift will vary from 25 to 100 feet. The apparatus is common in the East, and in 1890 the American consul at Cairo reported 20,000 of these in use in the upper and lower Nile valleys.

A form in which the buckets are attached to the wheel as in the old noria, but in which the wheel is driven by animal power as in the Persian wheel, is still used at times, and is known as the *Sakias*. The term *Taboot* is applied to such an apparatus when the buckets are replaced by bags made of animal skins.

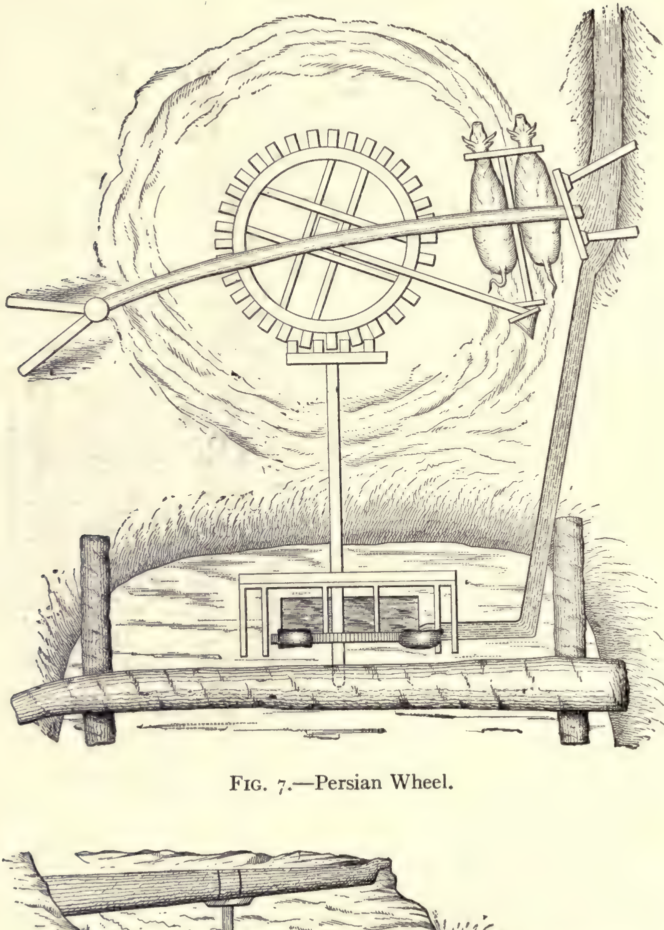


FIG. 7.—Persian Wheel.

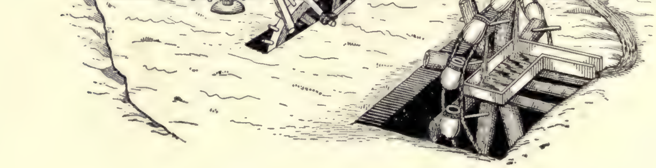


FIG. 8.—Persian Wheel.

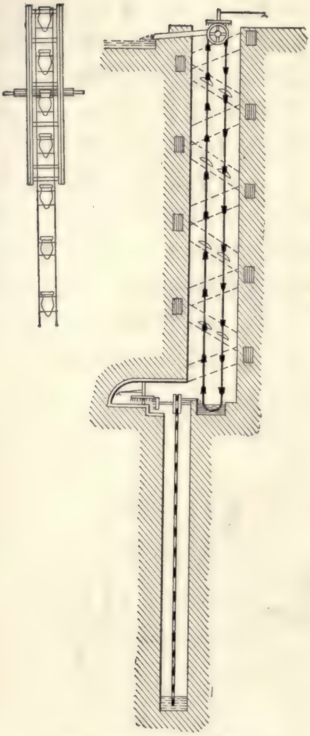


FIG. 9.—Joseph's Well.

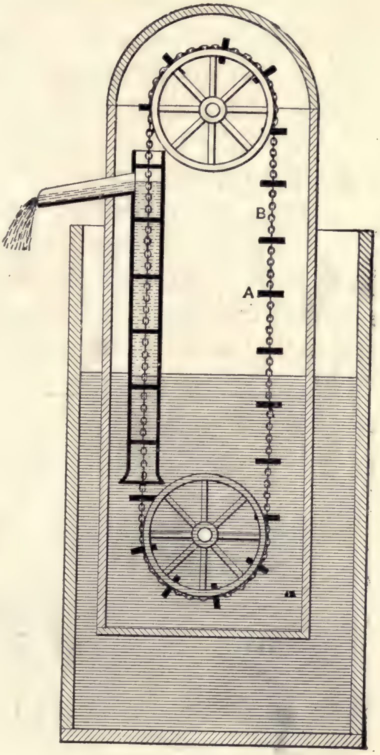


FIG. 10.—Chain Pump.

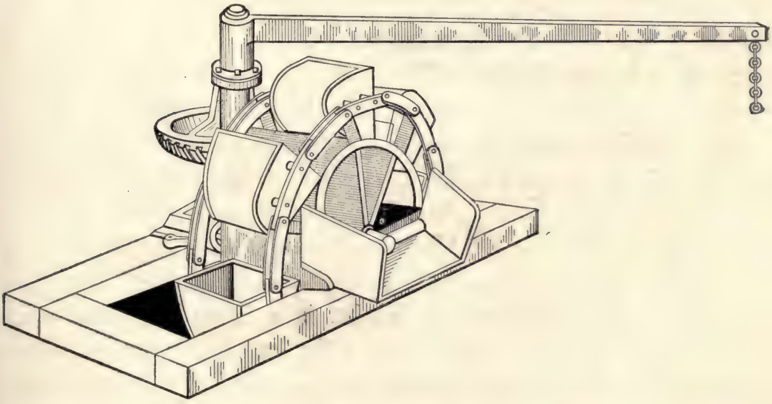


FIG. 11.—American Form of Persian Wheel.

An old Persian wheel whose date of construction is not known is located at Joseph's well at Cairo. This is shown in Fig. 9. The total depth is almost 300 feet, and this depth is divided into two lifts. The inclined passageway around the upper shaft enables the animals used for driving to be taken down to the lower driving wheel. The Romans built similar apparatus, calling them *Roman Buckets*. They were similar to our coal and grain elevators. Another development of this same apparatus consisted of a number of square discs mounted on a chain and fitted into a square pipe like small pistons. These chain discs, which have become the common chain

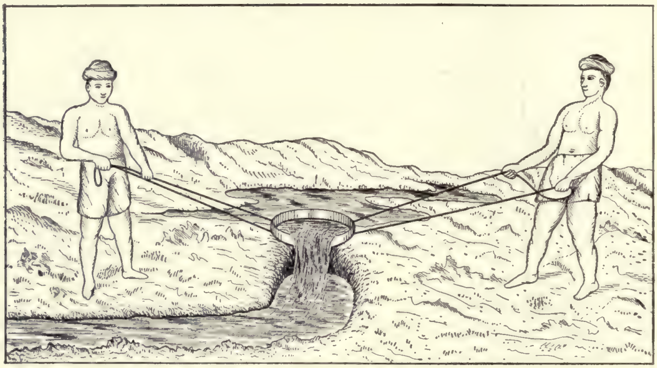


FIG. 12.—Mental.

pump of to-day, have been used in China from time immemorial. Fig. 10 shows the construction of this. The pallets or chaplets *AA* attached to the chain *B* are really loose-fitting pistons and in the application shown there is no suction.

The Persian wheel has been applied in America (Fig. 11); with it a horse is used and the crude gearing and buckets are replaced by metal parts. The principle, however, is the same as that of the old wheel. With such a machine a horse is said to lift 500 cubic feet per hour through a height of 20 feet.

The Mental, Katweh or **Latha** is a form of apparatus used in Egypt and India to-day (Fig. 12). It consists of a basket attached to two ropes in such a manner that two men may swing it into a stream when swinging it in one direction,

while on the return swing the basket by a dextrous twist is discharged. In this manner two men may raise 20,000 cubic feet of water one foot in a day of ten hours.

The Doon (Fig. 13) consists of a long trough pivoted near one end and balanced by a stone attached to an overhead lever. The operator stands on the long counterbalanced end of the trough, overcoming the counterbalance and causing the outer end of the trough to dip into the water. On stepping from this, the weight lifts the trough and discharges the water into the ditch above the stream or pond.

Another primitive apparatus was the **Mot**, which consisted

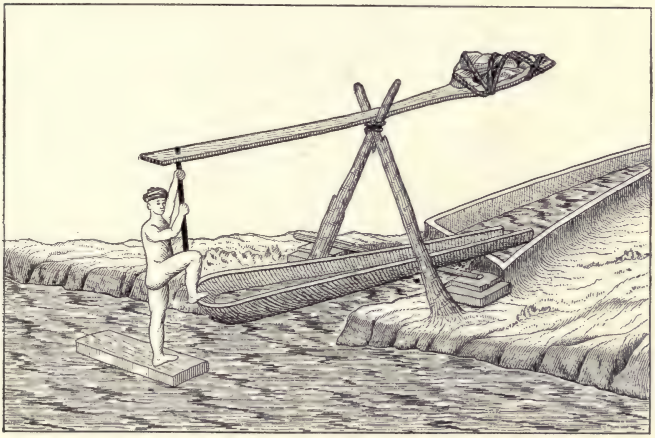


FIG. 13.—Doon.

of a bag of skin attached to a rope. The bag was raised to the surface by oxen, discharged, and then dropped back for another supply. This apparatus is effective, as two bullocks and a man can lift 79,000 cubic feet one foot in ten hours.

The double **Zig-Zag Balance** (Fig. 14) is shown by Mr. H. M. Wilson, as an apparatus used in Asia Minor and Egypt. Two men oscillate the frame and thus cause the water to flow past wooden valves at the intersection of the steps, and the water gradually passes from one step to the next. On each swing, water is lifted and gradually travels upward to the discharging trough.

Fig. 15 illustrates a method shown on certain Egyptian monuments. This should hardly be called a pumping appliance, but it serves to illustrate the great importance of irrigation when in ancient times such expensive methods were employed.

This brings one near the beginning of the Christian Era, and at this time there were several important applications of

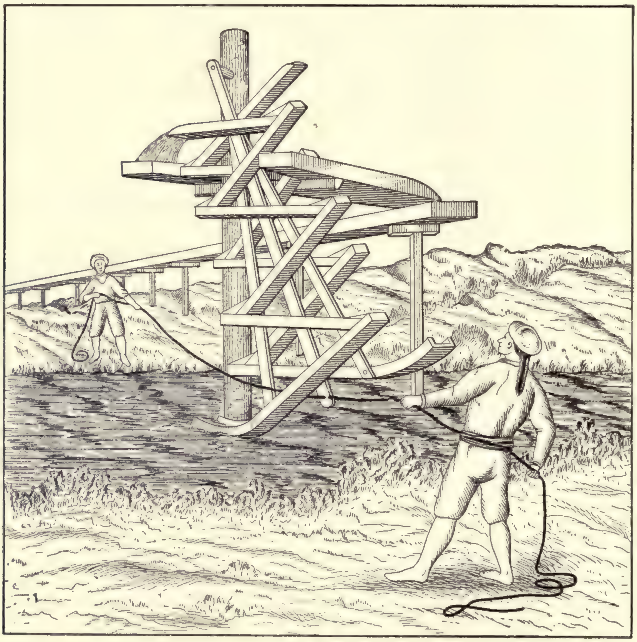


FIG. 14.—Zig-zag Balance.

mechanical principles to the raising of water. The use of suction to raise water was applied and valves were added to tubes carrying water while the movable diaphragm, partition or piston closely fitting the water vessel or tube was employed. The principle of the syphon was known at this time, and the syringe, which employed these various principles, was in use.

The principle of atmospheric pressure was not understood, although it was employed in the suction of water in early

machines. It was not until the time of Torricelli, 1644, that this was fully comprehended.

Fig. 16 is known as a **Tympanum.** It was employed

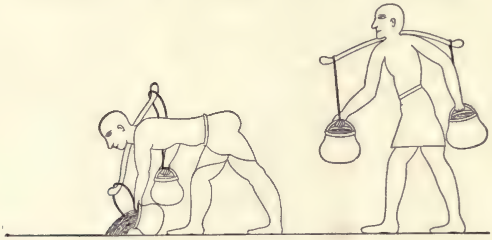


FIG. 15.—Egyptian Irrigation.

in Egypt. A spiral tube is attached to the face of the wheel, which is driven by the current. The end of the tube dips

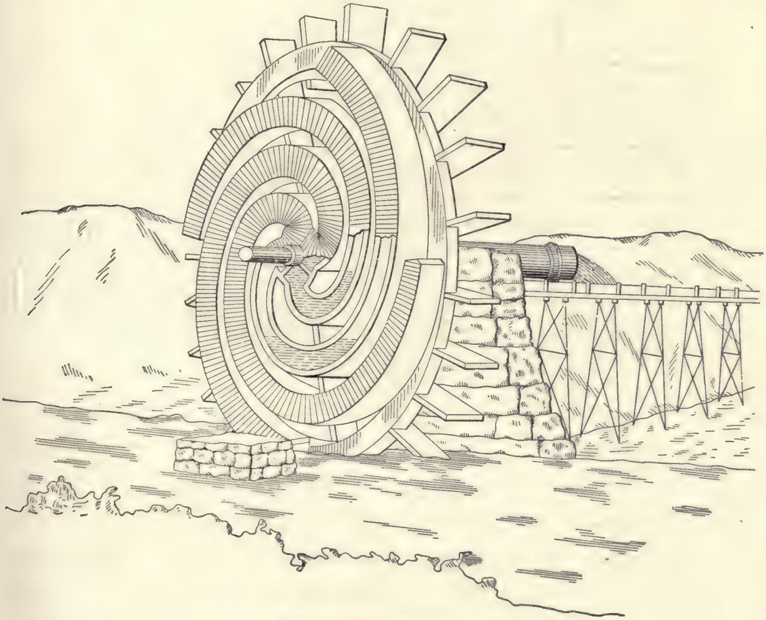


FIG. 16.—Tympanum.

beneath the stream, and as the wheel turns part of this water is caught within the tube and is gradually lifted to the hub of the wheel, where it is discharged into an irrigation flume.

The Archimedean Screw (Fig. 17) is one of the first combinations of a movable diaphragm in a tube. By turning the screw it is seen that the water is compelled to travel upward. The helical surface *A* on the axis *B* is rotated only, but the effect of it is that of axial movement.

The application of the principle of suction and one of the first uses of heat to lift water is described by Hero of Alexandria (*cir.* 120 B.C.) in his "Pneumatics." In this apparatus the heat from the burning sacrifice on the altar *A* (Fig. 18) is used to open the doors *B* of the temple. This was accomplished

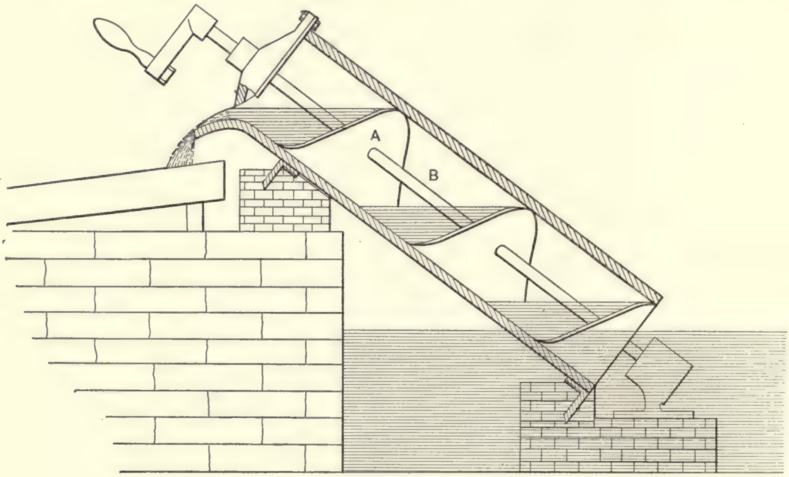


FIG. 17.—Archimedean Screw.

by the pressure produced on heating the air contained within the hollow altar *A*. The air was led into the top of the sphere *C* and its pressure drove the water into the bucket *D*, the weight of which acted on ropes attached to the trunnions *E* of the doors. When the sacrifice was burned the air contracted, producing a vacuum in *C*; the water was sucked back from *D* and the doors were closed by the weight *F*. In this apparatus it is important to notice the use of pressure to force water from one vessel to another, and of a vacuum or suction to draw water into a vessel.

From the description of Hero it is not possible to know

whether he invented any of the devices described, but it is reasonable to suppose that many of them were the inventions of Ctesibius, who made several mechanical inventions. The

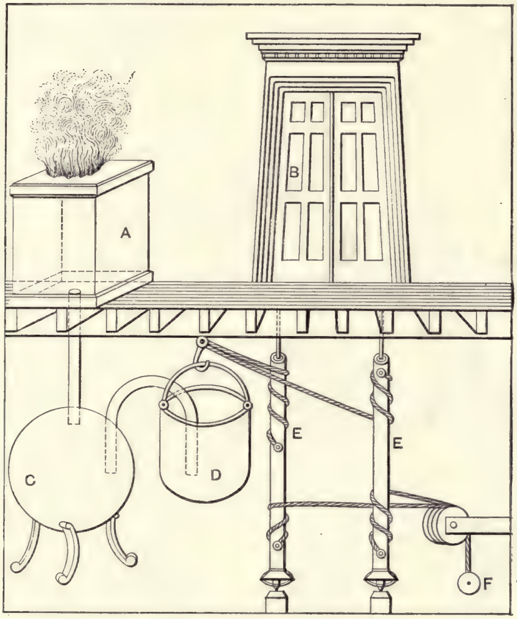


FIG. 18.—Temple Pump.

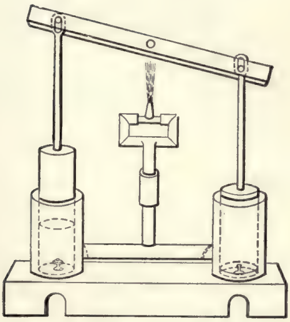


FIG. 19.—Force Pump.

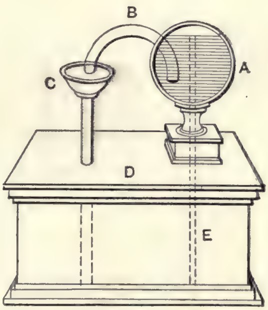


FIG. 20.—Sun Fountain.

Force Pump is ascribed to him and to Hero. This pump (Fig. 19) may have been suggested by the noria, in which valves were used at the bottom of the buckets to facilitate

filling. There were two single-acting pistons, moved up and down by a cross rod or beam, and although the pumps were single acting the stream was continuous, owing to the arrangement of the pumps. It is to be noted that this was intended for a fire pump, and is the earliest fire engine of which there is record.

Another machine (Fig. 20) called a **Fountain** uses the expansive force of heated air to operate it. Water is driven

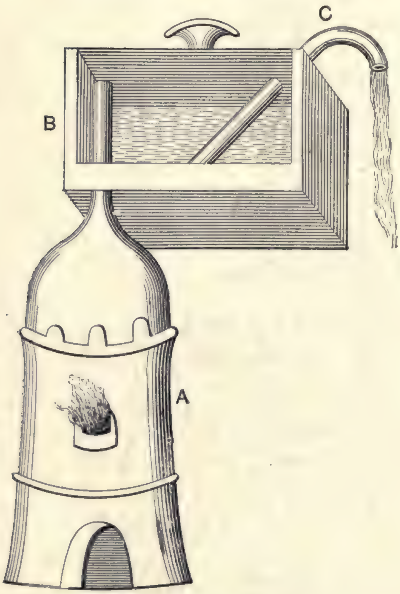


FIG. 21.—Steam Pump.

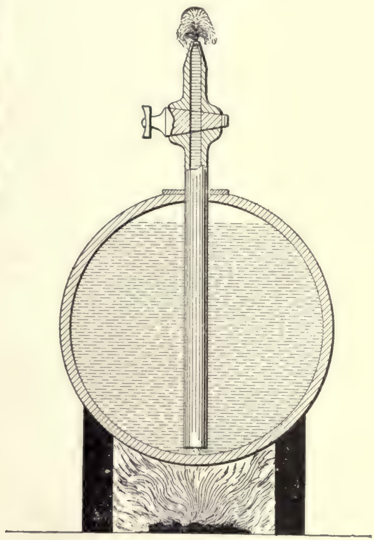


FIG. 22.—Fountain of de Caus.

from the vessel *A* through the syphon *B* by the expansion of the air in the top of *A* when exposed to the sun's rays. This water then falls into the box *D* and when the vessel *A* cools water is drawn up from *D* through the pipe *E*. Although Hero describes it in this manner, it is well to note that some form of valve must be placed in *C* and *E* to have the action proceed as described.

The use of steam pressure was suggested by Giovanni Baptista della Porta in 1601 in an apparatus (Fig. 21) in which

steam is generated in a boiler *A*, from which it is discharged into the top of a vessel *B* filled with water. A pipe *C* reaches to the bottom of *B*, and when the steam pressure is exerted on the water in *B* it is forced from the discharge pipe *C*. The apparatus is the same as that of Hero, and della Porta suggests in his book, published in 1601, that the vacuum caused by the condensation of the steam be used for filling the vessel with water. Della Porta should have shown another pipe leading from the vessel *B* to the source of water supply. It is interesting that della Porta suggested the separation of the boiler and

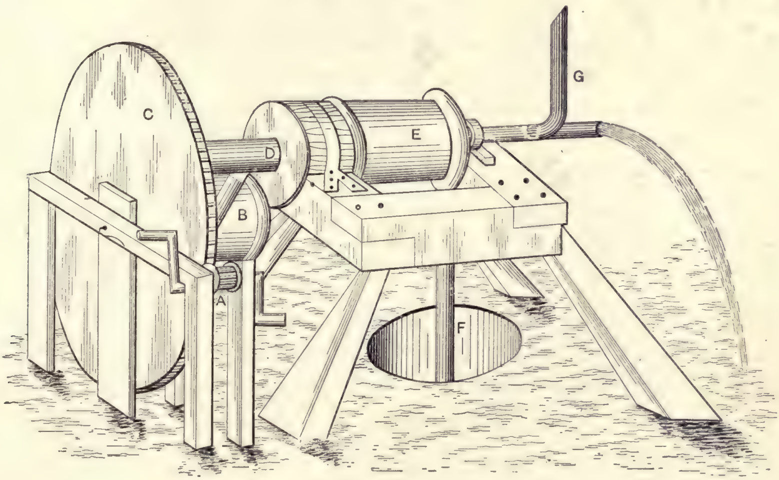


FIG. 23.—Pump of da Vinci.

the pump cylinder, a very valuable point, although in 1615 Solomon de Caus proposed to combine the two by placing the equivalent of the vessel *B* on the fire and heating the water in it for the purpose of driving the same by the pressure of the steam. (Fig. 22.)

Before this time, in the fifteenth century, Leonardo da Vinci suggested the pump shown in Fig. 23. The lantern wheel *A* was turned by two cranks and this motion was transmitted to a cylinder *B* through the toothed wheel *C*, and a helical groove in the cylinder *B* caused a piston rod *D* to travel back and forth by means of a pin on the rod which fitted into

the helical groove. The movement of the solid piston of the cylinder *E* would then suck water from the well through the suction pipe *F* and discharge it through the discharge pipe *G*. Valves must have been employed here, although not shown, and the machine at once suggests the complete understanding of the action of the piston within a cylinder.

In 1630 a patent was granted to David Ramseye by the English king which covered two points: "to raise water from low pitts by fire," and "to raise water from low places and mynes and coal pits, by a new waie never yet in use."

There is no record of what he did, but this seems to be one of the first useful applications of heat to the raising of water.

While the use of heat for pumping was progressing, the application of water power and animal power was being developed extensively. In England there was the installation of the London Bridge Water Works in 1581, by which the current of the Thames was used, while in France Agostino Ramelli published a book describing his inventions, including pumps. This book was published in 1588, and the pumps described were operated by men and animals as well as by the currents of streams. According to an old account the application of the lift pump was made in 1581, "when Peter Morrys was given a grant by the Lord Mayor and Commonalty of the city of London for the term of 500 years for supplying and conveyance of water into houses by pipes from an artificial force from London Bridge on condition that he pay ten shillings annually into the chamber of London." He was authorized to use the first arch of London Bridge for this purpose.

In a paper before the American Water Works Association, Mr. T. W. Yardley quotes from a description published in 1633 as follows: "The present supply of good water for London is like to be very much enlarged by the great improvement of the water works of Peter Morrys before mentioned, who, being a Dutchman, in the twenty-third year of Queen Elizabeth, first gave assurance of his skill in raising Thames water so high as should supply the upper parts of London; for the Mayor and Aldermen came down to observe the experiment, and

they saw him throw water over St. Magnus steeple, before which time no such thing was known in England as the raising of water." The success of this invention was such that Morrys obtained additional grants for two other arches under London Bridge. The second grant was for 2000 years and was finally secured by the New River Water Company.

In 1731 Henry Beighton, the engineer, described the London Bridge water works in the "Philosophical Transactions," and accompanied his minute account by an engraving. It may be that the pumping apparatus had changed from the time of its first installation, but that is not known.

There were three water wheels (Fig. 24) at the time of this

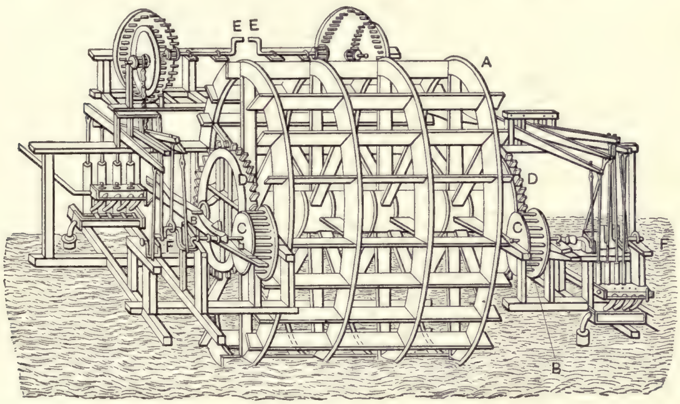


FIG. 24.—London Bridge Water Works.

description, each about 20 feet in diameter. These wheels *A* were carried on heavy frames *BB* and were raised and lowered with the tide. The frames *B* supporting the wheel *A* were pivoted at the axles of the lantern wheels *CC*, so that although the axle of the wheel *A* was raised and lowered the pin wheels *DD* attached to *A* on its axle were always in contact with the lantern wheels *CC*. The beams *BB* were supported at their outer ends by chains, which were raised or lowered by the cranks *EE* acting on a series of crown wheels and lantern wheels.

The axles of the lantern wheels *CC* were connected by a

coupling to crank shafts *FF*, each provided with four cranks, which in turn were connected to a series of vibrating levers. These levers were connected to pump rods at each end, although in the figure one set of pumps is omitted for clearness. There were sixteen pumps to the wheel with cranks arranged so that four of them would work alternately. The pump cylinders were 4 feet 9 inches long and 7 inches in diameter; they were fixed to the top of an iron cistern which contained the proper foot valves, while the discharge valves were placed in another box.

According to the description of this pump the gears were so arranged that the crank turned 2⅕ times per turn of the main wheel. With the several wheels used in 1733, containing 52 forcers or cylinders in all, 1954 hogsheads were pumped per hour while the main wheels made six turns per minute. This would occur with full tide, and at that rate 46,896 hogsheads would be pumped per twenty-four hours.

Beighton states that the amount of leakage or "slip" in the pump amounted to from 20 to 25 per cent of its displacement. The reasons for this he gives: 1st, the amount of valve lift will cause leakage; 2d, no leather can be made strong enough for pistons. He states that loose leathers cause leakage while tight leathers cause excessive friction.

Such pumps excite admiration when the amount of experience possessed at that day and the state of the art of both machinists and millwrights is considered.

While the works of Peter Morrys were being constructed in England, a book appeared in Paris, 1588, by Captain Agostino Ramelli, an Italian engineer. An account of this book on the various machines of Ramelli is given by Mr. W. F. Durfee in *Cassier's Magazine* for June, 1895. Among the machines for pumping water given by Mr. Durfee three have been taken to illustrate methods described by Ramelli, and also to illustrate the state of the art at this time. A number of the machines show that Ramelli was familiar with the action of the piston and the rotary engine as well as with the principles of gearing to transmit power between shafts at an angle and not in the

same plane; and for the purpose of obtaining reciprocating motion from continuous rotary motion.

In Fig. 25 a method is shown for draining an excavation or site beside a flowing stream. The wheel *A* is driven by the stream. Attached to its axle or shaft *B* is a series of cranks moving a set of levers *CC*, which cause the two shafts *D* to

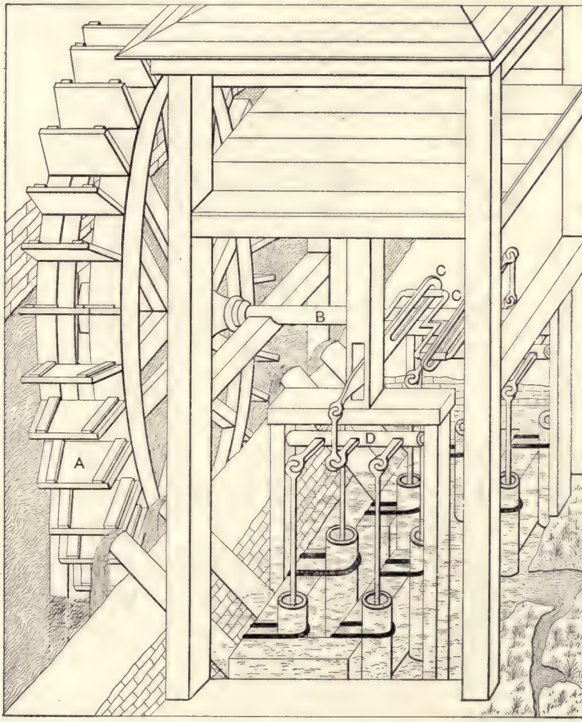


FIG. 25.—Ramelli's Pump.

oscillate. Each of the shafts *DD* has four arms which serve to drive the pump rods of four submerged pumps.

It is to be noted that in both this machine and that of Peter Morrys the crank and connecting rod are in evidence long before the time of James Watt. Here are also seen successful methods of securing reciprocating motion from rotary motion.

The wheel *A* of Fig. 26 was turned by a man walking on the inside, and in this the crank *B* gave a reciprocating motion to

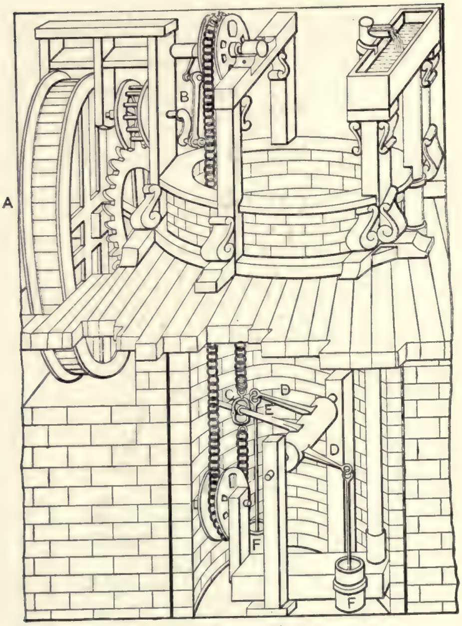


FIG. 26.—Ramelli Pump.

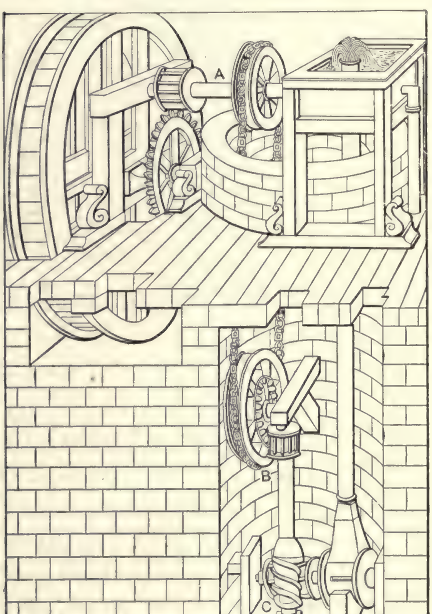


FIG. 27.—Ramelli's Rotary Pump.

the chain *CC*, which pulled in each direction on the arm *E*, driving the arms *DD* up and down and pumping on each stroke, one pump *F* forcing water on the up stroke of *E* and the other on the down stroke.

This not only shows the application of the crank and connecting rod, but there is much ingenuity displayed in arranging a chain to operate on both strokes.

The rotary pump (Fig. 27) of Ramelli is deserving of particular notice as a pump and also for the ingenious arrangement of its gears. For some reason the chain was not carried down to the axle of the pump. It may have been to keep this out of the water or for the purpose of getting a higher speed. The pin and lantern wheels at *A* and the crown and lantern wheels at *B* serve to increase the number of rotations, while the spiral gears at *C* serve only to change the direction of motion.

In 1628, two years before Ramseye secured his patent from King Charles for the use of fire, Edward Somerset, Marquis of Worcester, is thought to have installed an apparatus for raising water at Raglan Castle. He did not secure a patent on this until 1663, however, and there were no drawings nor even a model with his patent. From the description of the apparatus in his patent and from the grooves in the walls of Raglan Castle, an idea of the construction and operation of the machine may be formed, although it is not certain that this is correct in every detail.

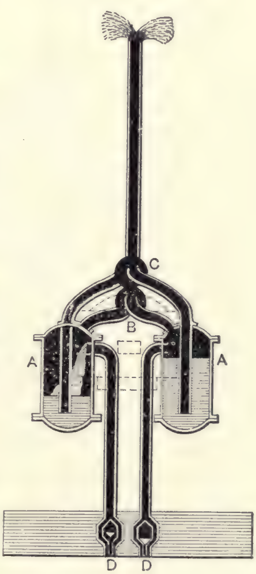


FIG. 28.—Worcester's Pump.

Two vessels *AA* (Fig. 28) are connected to a boiler through a valve *B*. In the figure the connection to the boiler is made to the right-hand vessel and the steam from the boiler presses on top of the water in *A* and forces it out through the valve *C*. During this action the right-hand foot valve *D* is held down on

its seat. The water is driven to a height corresponding to the steam pressure, and for that reason high steam pressure was required for great heads. While the right-hand vessel *A* is discharging, the condensation of steam in the left-hand vessel produces a vacuum in the vessel and water is drawn up through the foot valve *D*.

On the reversal of the valves *B* and *C* the operation just described is repeated with opposite vessels and so the operation became continuous.

Worcester introduced this for the water supply at Vauxhall near the old city of London. Upon examination it is seen that apart from the useful application of the invention there is nothing new in this apparatus, as the elements are all found in della Porta's work.

In 1683 Sir Samuel Moreland, the master mechanic of the laboratory of Charles II, published a book from Paris on "The Elevation of Water by All Sorts of Machines." He had been sent to Paris by Charles on business relating to the water works which the king had erected. In his book Moreland speaks of the expansion of steam and its pressure, showing that a good idea of the pressure and volume of saturated steam was common in that day. He also refers to the duty of his pumping engines, using the term as it is used to-day—the amount of work per hundred weight of coal. The question of the application of steam to the raising of water was one which not only occupied Moreland, but many other mechanicians, on account of the difficulty which was then experienced in clearing the shafts of the English mines from the vast quantities of water which collected therein from the underground flow. Many mines had to be abandoned because of the cost of carrying the mines deeper when this expense of draining was too great. The work was mostly done by animal power and the cost was rather startling when compared with the steam pump of even the early days.

It is to be noted that Moreland introduced and invented the plunger type of pump in 1675. This type of pump was one in which an enlarged end of a rod was forced into a cham-

ber, displacing the water. This was followed shortly afterward by a bucket pump, in which the water passed through valves in the piston on the down stroke, as was the case in the later engines of Simpson used at Thames-Ditton.

In 1698 Thomas Savery patented the design of an engine

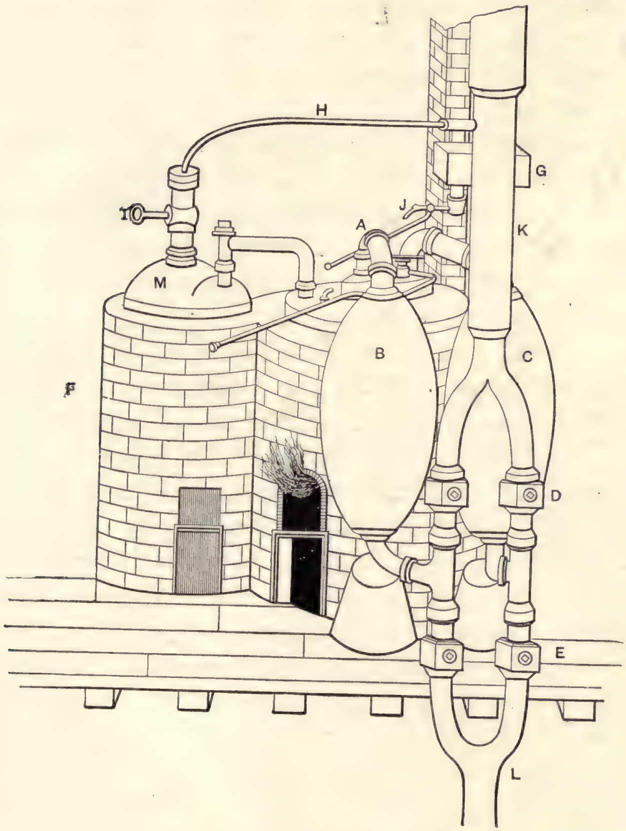


FIG. 29.—Savery's Pump of 1702.

for freeing the mines of Cornwall from water. It was the first steam apparatus applied to this kind of work. In 1699 he submitted a model to the Royal Society of London and successful experiments were made with it. A model fire engine was exbihited before King William III at Hampton Court in 1698, and the success of this led to the granting of the patent which

read: "A grant to Thomas Savery, Gent'l., of the sole exercise of a new invention by him invented for raising of water and occasioning motion to all sorts of mill works by the impellant force of fire, which will be of great use for draining mines, serving towns with water and for the working of all sorts of mills, when they have not the benefit of water nor constant winds; to hold for 14 years with usual clauses."

The apparatus is almost identical with that of Worcester, and it is not known whether or not Savery knew of the earlier work. The form of his pump of 1702, which is an improvement on that of 1698, is shown in Fig. 29. It consists of a main boiler A and the pumping chambers B and C. The water from the tank G discharges through the valve J on one of the pumping vessels, which condenses the steam in that vessel and the vacuum produced thereby draws water through the suction pipe L and foot valve E. When steam is then admitted from A through the steam valve, the water is forced out through the valve D into the discharge pipe. When the water is low in the boiler A, the auxiliary boiler is filled from the main K through the pipe H and valve I. This water is driven into the main boiler by raising steam in M. The pipe connecting the two boilers is carried to the bottom of the auxiliary boiler M and the steam pressure on the water drives it over into the main boiler A, provided the pressure in M is higher than that in A.

Savery seems to have been a wide-awake promoter and advertiser, for he began a systematic scheme for making his invention known. He explained it to the Royal Society and presented them with a drawing and specifications which appeared in their "Transactions," and he published a prospectus, called "The Miner's Friend; or a Description of an Engine to raise water by fire described and the manner of fixing it in mines, with an account of several uses it is applicable to, and an answer to the objections against it. London, 1702." This invention of Savery was intended to do away with the great expense in the use of animal power to operate the pumps of the mines, which not only were expensive, but reached their limit of capacity, so that workings could not be carried much

farther. In one mine 500 horses were employed in handling the water.

Savery's improvements were the addition of surface condensation, the secondary boiler, and the use of water cocks. It must be remembered that this machine could not be used in deep mines, as sufficient steam pressure could not be carried. The joints of the sheets forming the boilers, pump chambers, and other parts were fastened by solder, and at high tempera-

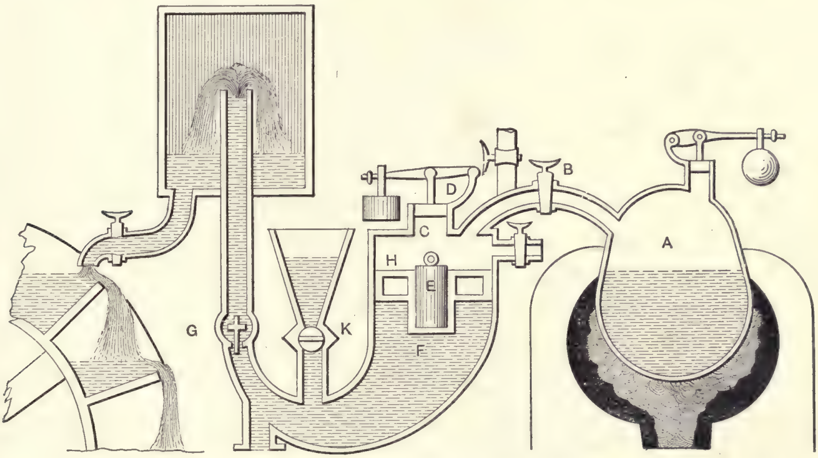


FIG. 30.—Piston Pump of Papin.

tures these joints would not hold. Desaguliers reports such accidents.

The Savery engine was built by others after his time—by such engineers as Desaguliers and Gravesande.

While Savery was using the direct action of steam for raising water, Papin, in Germany, was proposing the use of the piston to separate the steam and the water, thus leading to the further application of two pistons of different sizes, so that the steam pressure did not have to be equal to the water pressure. In 1707 Papin proposed a water pump shown in Fig. 30. This was brought out in Cassel, Germany. In this machine steam was generated in the boiler *A* and was conducted through the cock *B* to the cylinder *C*. It acted on the piston *H* and forced water

through the discharge valve *G* into an air chamber, from which it was delivered. Water was admitted through the valve *K* after the piston had been driven to the end of its stroke. Papin realized the cooling effect of the cylinder walls and suggested the introduction of a piece of heated iron *E* to warm up the piston before the introduction of the steam. The weighted cover *D*, similar to Papin's safety valve, served for the introduction of the iron.

Although Papin had in his pump the possibility of making the steam pressure different from that of the water, it was not to him that the honor for the first use of pistons of various sizes is due. Pistons for the forcing of water were explained by Ramelli and Leonardo da Vinci, but these pistons were driven through some mechanical medium from water power or man power, and it was Thomas Newcomen, a blacksmith of Dartmouth, England, who first employed steam for the operation of a water piston of different size from the steam piston. This at once enabled mines to be sunk to a greater depth, as pumping could now be done with greater efficiency. The Savery engine was used to lift water 350 feet, but with the engine invented by Newcomen the height was limited only by the strength of the materials employed.

The engine of 1705 is shown in Fig. 31. Steam is generated in boiler *A* at about atmospheric pressure, and as the piston of the cylinder *B* is drawn up by the unbalanced weight of the pump rod *C* steam is drawn into the cylinder from the boiler through a valve at the top. When the piston reaches the top of its stroke the valve is closed and water is sprayed into the steam space from reservoir *D*, thus condensing the steam in the cylinder and producing a vacuum. The air pressure on top of the piston is then sufficient to force the piston down, raising the pump rod *C*, and with it the pump piston at its lower end. The water of condensation then falls through a pipe *G* into a hot well, the height of the water in the drain being fixed by the vacuum in the cylinder. The walking-beam *E* served to connect the two piston rods by chains and sectors, and the rod *C* could be of any length, as the greater part of its

weight and that of the column of water in the discharge pipe could be balanced by counterweights, on a rod such as *F* on either side of the center. Sufficient weight was left unbalanced to cause the piston of cylinder *B* to rise when low-pressure steam was admitted.

Since the engine was really driven by atmospheric pressure and was operated by steam at practically no pressure above the atmosphere, it was known as "the atmospheric engine."

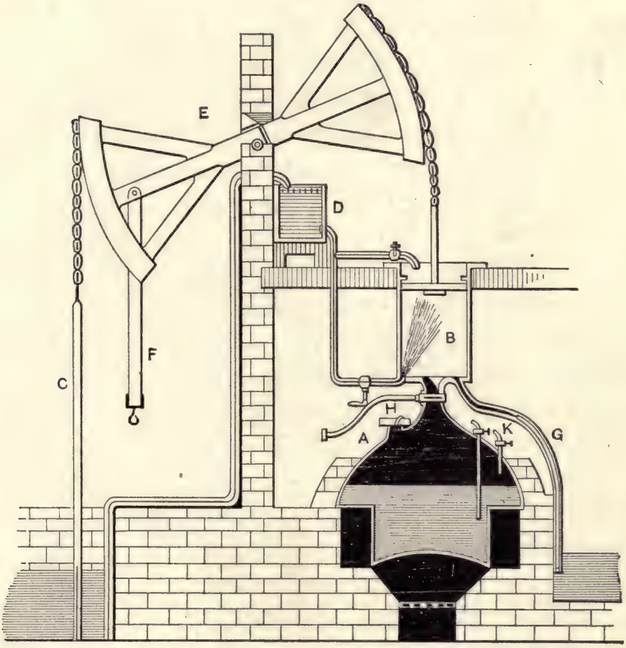


FIG. 31.—The Atmospheric Engine of Newcomen.

Indeed, so low was the steam pressure that the weight of the safety valve *H* was sufficient to keep it closed.

The try cocks *K* served to indicate the quantity of water in the boiler, and the cock above the cylinder *B* was used to introduce water on the top of the piston to keep the edge of it airtight. The use of a spray inside the cylinder to condense the steam resulted from the discovery that leakage of water around the piston condensed the steam more quickly and

better than the original method of cooling the cylinder wall by spraying it with water.

It is remarkable that a man of such little training as Newcomen appears to have had should have been able to combine the necessary elements to make such a great machine. He did not occupy a very high position in the town; however, he was a good workman. When he and his colleague, John Calley, wrote to Dr. Hooke, the famous physicist, in regard to their use of a steam cylinder and piston to drive a separate pump, he advised against their plan. They were not to be put down, however, and in 1705 they secured a patent.

The engine was then applied to drain mines and pump water, but Newcomen and Calley did not have sufficient mathematical knowledge to properly design their machines and they had many failures, while their successes were accidents.

In 1713 Humphrey Potter, a boy who operated valves by hand, arranged an automatic method of shutting off the steam and water by the use of beams, strings, and catches, and made the machine independent of an attendant. Henry Beighton improved this in 1718 by substituting a vertical beam with pins which struck the valve handle as it was raised and lowered by the walking beam. The vertical beam was known as a "plug rod," "plug tree," or "plug frame."

These pumps were built for various purposes and were of different dimensions; one in 1714 for Ansthorpe in Yorkshire was 23 inches in diameter and of a 6-foot stroke. It made 15 strokes per minute. One, described by Farey, was 8 inches in diameter on the water side and 24 inches on the steam side. The stroke was 60 inches and there were 15 strokes per minute. This pump lifted water 162 feet and the water column on the piston weighed over 3500 pounds, which with 660 pounds of unbalanced weight of the pump rod necessitated almost 5000 pounds on the piston. Such a pressure could be obtained with a vacuum of 21 inches of mercury. The pump developed 8 horse power. Another engine at Griff in Warwickshire cost £150 per year to operate it and displaced 500 horses at an expense of £900 per year. The first Newcomen engine

was introduced on the continent in 1723 at Königsberg, Hungary. In 1735 cast iron was used in place of wrought iron for the parts of the engine.

The engine was improved by many engineers. About 1769 John Smeaton, one of the most distinguished engineers of his day, built several engines with greater strokes than those usually employed. By using the proper diameters for his pistons he was enabled to get much higher speed. Before building pumps he experimentally determined the proper proportions of the engine and so improved its construction.

Before the last quarter of the century these engines were introduced to such an extent that the coal mines of Coventry and Newcastle, the tin and copper mines of Cornwall, blowing engines of the English and Scotch furnaces, the docks of Cronstadt in Russia, the lowlands of Holland and the salt mines of Hungary bore testimony to the success of this invention. The mines were carried to greater depths, the cost of pumping water and air was reduced, and the supply of water to towns was more certain.

One of Smeaton's Newcomen engines is shown in Fig. 32. The figure shows the cylinder *A* connected with the boiler by means of the steam pipe *B*. The boiler is placed in another building. The valve *C* admits steam to the cylinder through the admission pipe, which is carried above the bottom of the cylinder so as to keep the injection water from entering it. When the steam is admitted, the piston is driven or pulled upward and when the top of its stroke is reached the upward movement of the plug tree or working plug *D* acts on the handles *E* through pins, turning the axle *F*, and the Y or "tumbling bob" *G* is thus moved, shifting the rod *H* and handle *I* and thus shutting off the steam. At the same time the handle *K* opens the valve *M*, allowing water from the cistern *N* to enter the cylinder through the spray head *O*. This immediately condenses the steam in the cylinder and the vacuum produced permits the atmospheric pressure to drive down the piston. The pins *P* and the springs *Q* stop the downward motion at the proper point. At this lowest point the

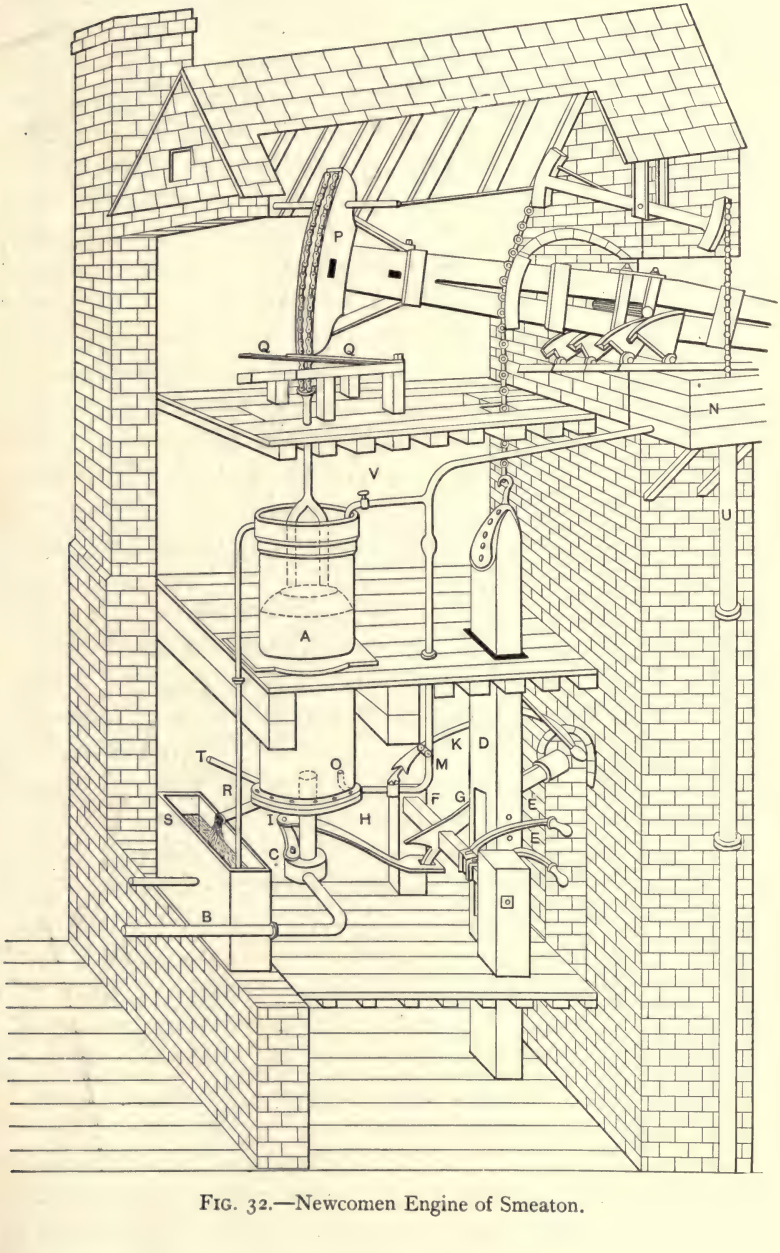


FIG. 32.—Newcomen Engine of Smeaton.

drain pipe *R* is opened, allowing the condensed steam and condensing water to discharge into the hot well *S*. The sniffing valve *T* is then opened, allowing any air to escape. The feed water of the boiler is taken from the hot well. The cistern *N* is supplied by a jack-head pump *U*, driven from a small beam. Both beams get their motion through chains and sectors, so that there is always a straight pull on the piston or pump rods. The cock *V* admits water around the piston so that the oakum packing of the piston is kept in proper condition. The excess of this water is carried off through the drain pipe to the hot well. The main beam is carried on sectors so as to reduce the friction.

To improve the efficiency, Smeaton covered the steam side of the piston with planks and when the injection water contained salts which would form a scale, the water in the hot well was not used for boiler feed, but the clear feed water was passed through a coil of pipe immersed in the hot well.

The next important step in the improvement of the steam engine and pump was that of James Watt. The invention of this man was one of the greatest events in the history of civilization, as it not only improved the existing machines, but in his specifications are contained the fundamental ideas of all modern improvements to the steam engine. This event clearly demonstrates what may be done by a careful and detailed study of existing conditions.

The early history of Watt, who was born in 1736, is one with which every engineer should be familiar. After many trials and successes in the south of England he came back to Glasgow and was employed to repair some of the apparatus belonging to the university. In 1763 he repaired its model of the Newcomen engine, and this led to his making a study of the history of the steam engine. He read the treatise of Desaguliers and the works of others. In this study he learned of the accomplishment of Savery, Newcomen, and those who had preceded them. Watt now began a series of experiments on the action of the engine, determining quantitative relations between the

amounts of steam, cooling water, and the heat of steam and water. He discovered the great loss by radiation and absorption and was led to use non-conducting materials for his vessels as well as for the coverings of them. Not having experimental data for this work, he made a series of original experiments on the temperature and pressure of steam at whatever points he could observe these and constructed a curve to give values at other points. He determined the amount of steam used by the Newcomen engine and the amount which should have been used had none of the steam condensed; in addition he compared the amount of injection water used in the engine with the amount which should have been used. These experiments showed him that three-fourths of the steam taken into the cylinder was wasted and that the engine used four times as much injection water as it should have used. His calculation showed him at once that the method of producing the vacuum was a poor one, as the cylinder had to be heated by steam at each stroke so that it could be filled, and then this heat was removed again on the condensation of the steam. He then tried to keep the cylinder hot, which necessitated that the steam be taken from it for condensation. He invented the independent condenser, which made his improvement complete. After constructing a number of experimental machines and after many vicissitudes he took out his patent in 1769 in connection with Dr. Roebuck.

The patent of 1769 gave the following description:

"My method of lessening the consumption of steam, and consequently fuel, in fire engines, consists in the following principles:

"1st. That the vessel in which the powers of steam are to be employed to work the engine, which is called 'the cylinder' in common fire engines, and which I call 'the steam vessel'—must, during the whole time that the engine is at work, be kept as hot as the steam which enters it; first, by inclosing it in a case of wood, or any other materials that transmit heat slowly; secondly, by surrounding it with steam or other heated bodies; and thirdly, by suffering neither water nor other sub-

stances colder than the steam to enter or touch it during that time.

"2dly. In engines that are to be worked, wholly or partially by condensation of steam, the steam is to be condensed in vessels distinct from the steam vessel or cylinder, though occasionally communicating with them. These vessels I call condensers; and while the engines are working, those condensers ought at least to be kept as cold as the air in the neighborhood of the engines, by application of water or other cold bodies.

"3dly. Whatever air or other elastic vapor is not condensed by the cold of the condenser, and may impede the working of the engine, is to be drawn out of the steam vessels or condensers by means of pumps, wrought by engines themselves, or otherwise.

"4thly. I intend in many cases to employ the expansive force of steam to press on the pistons or whatever may be used instead of them, in the same manner as the pressure of the atmosphere is now employed in common fire engines. In cases where cold water cannot be had in plenty, the engines may be wrought by this force of steam only, by discharging the steam into the open air after it has done its office.

"5thly. Where motions round an axis are required, I make the steam vessels in form of hollow rings or circular channels, with proper inlets and outlets for the steam, mounted on horizontal axles like the wheels of a water mill. Within them are placed a number of valves that suffer any body to go round the channel in one direction only. In these steam vessels are placed weights, so fitted to them as to fill up a part or portion of their channels, yet rendered capable of moving freely in them by the means hereinafter mentioned or specified. When the steam is admitted in these engines between these weights and valves, it acts equally on both, so as to raise the weight on one side of the wheel, and by the reaction of the valves successively, to give a circular motion to the wheel, the valves opening in the direction in which the weights are pressed, but not in the contrary. As the vessel moves round, it is supplied

with steam from the boiler, and that which has performed its office may either be discharged by means of condensers, or into the open air.

"6thly. I intend in some cases to apply a degree of cold not capable of reducing the steam to water, but of contracting it considerably, so that the engines shall be worked by the alternate expansion and contraction of the steam.

"Lastly, instead of using water to render the piston or other parts of the engine air- or steam-tight, I employ oils, wax, resinous bodies, fat of animals, quicksilver, and other metals in their fluid state."

It is to be noted that these claims covered the following points:

1st. Lagging and jackets.
2d. Condensers.
3d. Air pumps.
4th. Expansive use of steam and the non-condensing engine.
5th. A rotary engine.
6th. Packings.

Mathew Boulton became the partner of James Watt, and it is to him that much of the credit of the actual engine is due. He was the owner of one of the most famous manufactories of the day at Soho, near Birmingham, England. Here he manufactured ornamental metal ware, gold- and silver-plated ware and works of art, such as vases, statues, and bronzes. His factories were noted for the good work done, and for the broad policy of management.

Although the arrangement of the partnership was agreed on in 1769, it was not until the spring of 1774 that Watt could go to Birmingham. By November of that year their first engine was built. The form of this is shown in Fig. 33.

Steam enters from the boiler by the pipe A and the valve B passing to the steam jacket C. The condenser D is then connected to the cylinder by the valve E, and the vacuum thus produced in the space F causes the piston G to move downward, and steam flows in above the piston. When the piston reaches the lower end of the stroke, the valves B and E

are closed while the valve *H* opens. This connects the spaces on each side of the piston, and the weights of the pump rods *I* and *J* on the outer end of the beam *K* overbalance the weight of the piston *G* and its rod, and so the piston is pulled rapidly

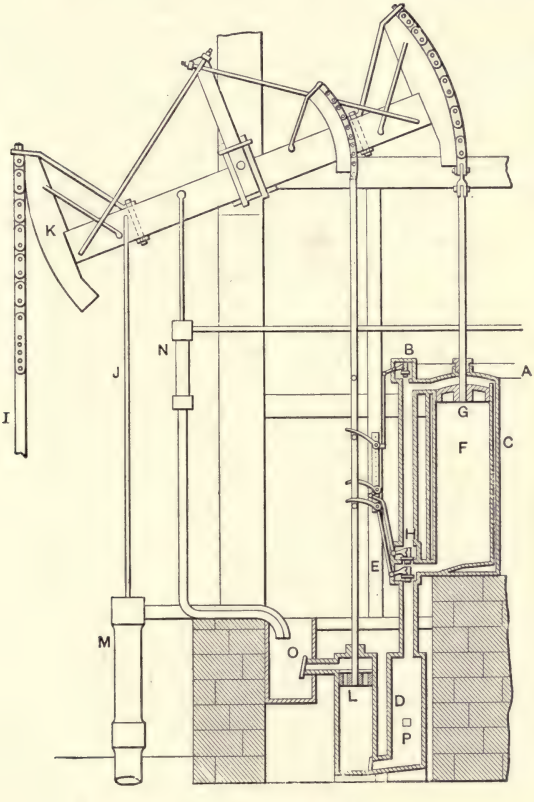


FIG. 33.—Watt's Engine.

upward to the top of the cylinder, the steam above the piston passing over to the lower side.

After closing *H* and opening *B* and *E* the operation is repeated, and the air pump *L* removes the condensed steam and the air from the surface condenser. The pump *M* supplies the cooling water, and the pump *N* takes water from the hot well *O* and feeds the boiler. The pump rod of the air pump contains the pins which operate the handles of the valves.

An outlet P is used when the air is driven out from the cylinder and the air pump before the engine is started.

There was much trouble in getting men and machines to make these engines, and it may be said that the demand for better work developed the machinist's trade of that day. Much of the development was made by the firm of Boulton & Watt.

In the building, erection, and operation of their engines, Boulton & Watt were led to take out patents for the following articles:

A letter copy press.

A cloth dryer by the use of steam in copper rolls.

Five devices for getting rotary motion from reciprocating motion without the use of a crank.

The expansive use of steam.

The double-acting engine.

The double-coupled engine.

A rotary engine.

A trunk engine.

A steam hammer.

Parallel motion.

The engine governor.

Mercury steam gauge.

Water gauge.

Steam-engine indicator.

Watt seemed to be one who could always find some means of meeting every need: when it took too much time in copying his reports to Boulton, he invented the copy press; when it was necessary to study the action of steam in the cylinder, he brought out his indicator. These numerous inventions do not indicate that the firm was always successful. Many times they were on the verge of bankruptcy, and had their patent not been extended for twenty-four years, when it first expired, their labor would have been in vain, because of financial straits. The extension gave them the needed relief, and at the expiration of the patent the firm was in good condition. The story of the trials and successes of this firm in the development of the engine is given in the biographies

of these two men, and the student is recommended to study these most interesting books.

To gauge the power of his pumps, Watt introduced the term "horse power," in so common use to-day. This, with the term, "duty," gave those using pumps a method of comparing

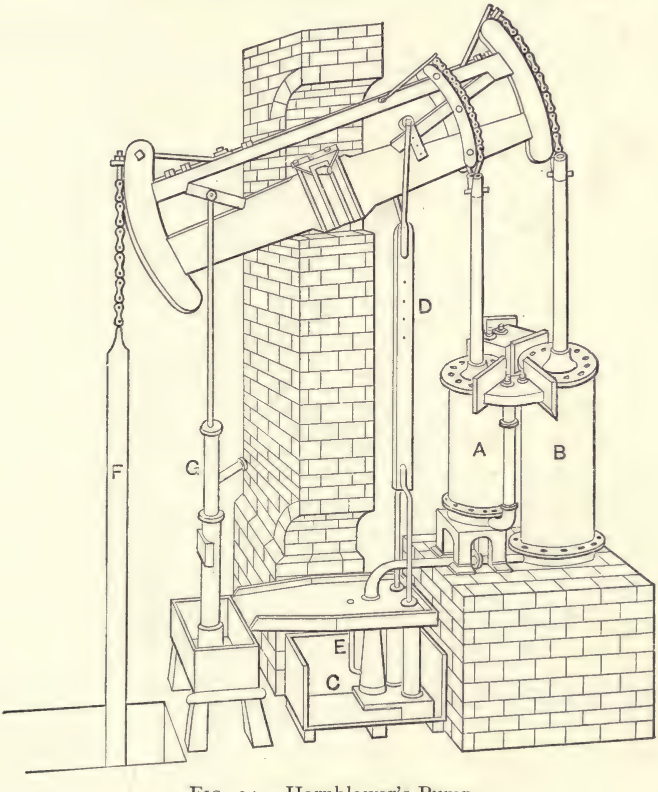


Fig. 34.—Hornblower's Pump.

the operations of various machines. An economical operation was aimed at in all of this work, and in order to interest the engine attendants, monthly prizes were given for the best results in duty during the month. An operative machine had been constructed, and it was now their object to improve the efficiency of it.

Boulton died in 1809 and Watt in 1819, but before that

time these men had given the business over to their sons. They enjoyed the protection of their fathers' patents until 1801, during which time others were at work on the improvement of the engine end of the pump.

Jonathan Hornblower patented a compound engine in 1781, although Watt claimed this invention. The engine is shown in Fig. 34. Steam is admitted into the cylinder *A* from the boiler, and from this cylinder it discharges into the cylinder *B* and thence into the condenser *C*, shown in section. The plug-tree rod *D* serves as the rod for the air pumps as well as to operate the valve handles, which are not shown. The operation of the engine is practically the same as that of the Watt engine. To start, all air is driven out by allowing boiler steam to flow through the cylinders and condenser, and thence through the sniffing valve at *E*. The condenser will now condense steam below the piston in *B*, and as the piston descends the valve between *A* and *B* allows steam below the piston of *A* to expand and press down on top of the piston of *B*. The steam from the boiler enters on top of that of *A*. When the bottom of the stroke is reached the boiler and condenser are cut off and the top of each cylinder is connected to the lower portion. The weight of the main pump rod *F* and the boiler feed pump *G* pulls the pistons upward and the operation is repeated. This was declared an infringement on the Watt patent. It did not give a much higher duty than the best single-cylinder Watt engines of the day, although the same idea as applied by Arthur Wolf in 1804 with higher pressure steam gave duties of from 40,000,000 to 57,000,000 foot-pounds per bushel of coal, while the Watt engine gave a little over 30,000,000.

The Bull Cornish pumping engine of 1798 was brought out by William Bull and Richard Trevithick. This type of engine is the one which remained in use longer than any other, as it was much simpler than that of Watt and had all of the elements of economy. It is shown in Fig. 35.

The steam cylinder *A* is carried on the timbers *BB*, extending from the walls of the pump house in such a manner as to bring the piston rod directly over the pump well. The piston rod

C is connected to the pump rod *D*, and this, in turn, to the counter balancing beam *E* by the rod *K'*. A pump rod *G* with its pump *F* is also shown. The counterweight *H* is to balance as much of the weight as is thought necessary. The rod *I*

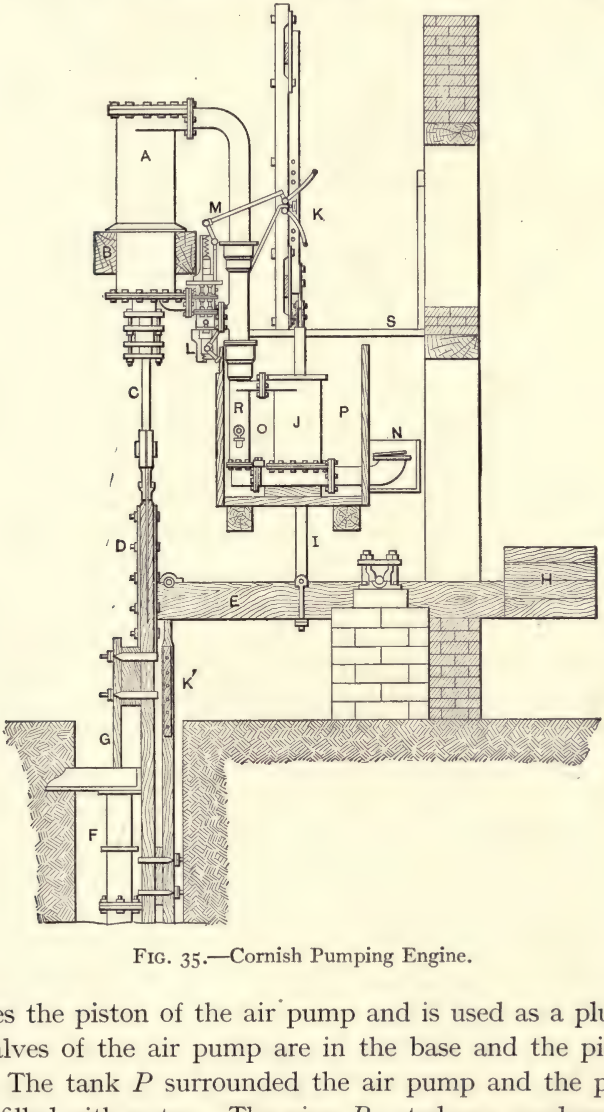


FIG. 35.—Cornish Pumping Engine.

actuates the piston of the air pump and is used as a plug rod. The valves of the air pump are in the base and the piston is solid. The tank *P* surrounded the air pump and the pipe *R*; it was filled with water. The pipe *R* acted as a condenser, but

water was admitted through O, making it really a jet condenser. The pins on the plug rod K operated the rods leading to the valves at L and M. In starting, the valves were operated from the floor S and the air was driven out from the cylinder, condenser, and air pump through the sniffing valve N, which was water-sealed.

This engine was adjudged an infringement on the Watt patents, which prevented its introduction for some time, but it was afterward used exclusively in America and Europe with many improvements. It had many advantages over the other engine in its simplicity, but it was objectionable in that it must be placed directly over the opening of the mines. It was of great value further in that by properly selecting the

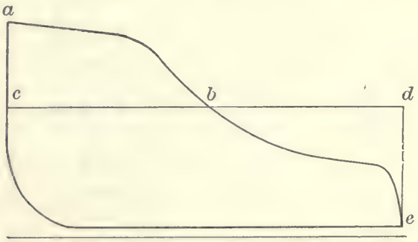


FIG. 36.—Action of Cornish Engine.

mass of the parts the inertia of these could be used to permit the expansion of the steam, although the water pressure or resistance was constant. This may be shown by the diagrams of Fig. 36. The steam pressure in excess of the resistance of the water is used in accelerating the piston, piston rod, pump rod, balancing beam, and counterbalance, and after passing the point at which the steam pressure equals the water pressure the inertia of the parts will supply the deficiency of energy on being brought to rest after the steam pressure falls below the resistance of the water. Neglecting friction the area *abc* will equal the area *bde*. These areas will always be a measure of the energy stored up in the moving parts and will be a function of the maximum velocity and the mass moved, so that by changing the amount of mass the speed of the apparatus

could be altered. The use of heavy counterweights at times required all of the steam pressure on the up stroke of the weight to move them, while on the down stroke the excess of counterweight acting with the steam was used to lift the water from deep mines.

It is important to note the advantage of use of steam expansively, as it is this which has made the modern pumping engine with a fly wheel so economical. These early engines were quite efficient, as is seen from the table below, which demonstrates the great advantage of this Cornish engine.

DUTIES IN FOOT-POUNDS PER BUSHEL (94 POUNDS) OF WELSH COAL

In 1769, the Newcomen engine	5,500,000	ft.-lbs.
In 1772, the Newcomen engine, improved by Smeaton	9,500,000	"
In 1778 to 1815, Watt engine	20,000,000	"
In 1820, improved Cornish engine (average)	28,000,000	"
In 1826, " "	30,000,000	"
In 1827, " "	32,000,000	"
In 1828, " "	37,000,000	"
In 1830, " "	43,350,000	"
In 1839, " "	54,000,000	"
In 1850, " "	60,000,000	"
In 1827, highest duty, Consolidated Mines	67,000,000	"
In 1832, " Fowey Consols	97,000,000	"
In 1842, " United Mines	108,000,000	"

This engine was developed into the beam engine and was used for water works. Fig. 37 is a cut of a Cornish engine of 1840 for the East London Water Works, with a capacity of 6,000,000 gallons per twenty-four hours. This represented the best engine of the day.

While the Cornish engine was being used for pumping water from mines and for the water supply of cities, another form of pump was sucessfully operated in 1830 in New York by a Mr. McCarty. This was the **centrifugal pump.** It was a pump which had been known for a long time, as Euler discussed its theory in a paper in 1754. According to one author the invention of it is due to Denys Papin in 1689, who took his idea from Johann Jordan. Jordan designed a centrifugal pump in 1680. Demour in 1730 invented the equivalent of a centrifugal pump. It

consisted of a tube, Fig. 38, mounted on a vertical axis so that the lower end entered the water to be raised near the axis. On turning this the centrifugal force overcame the effect of gravity and the water rose. In 1818, a few years before

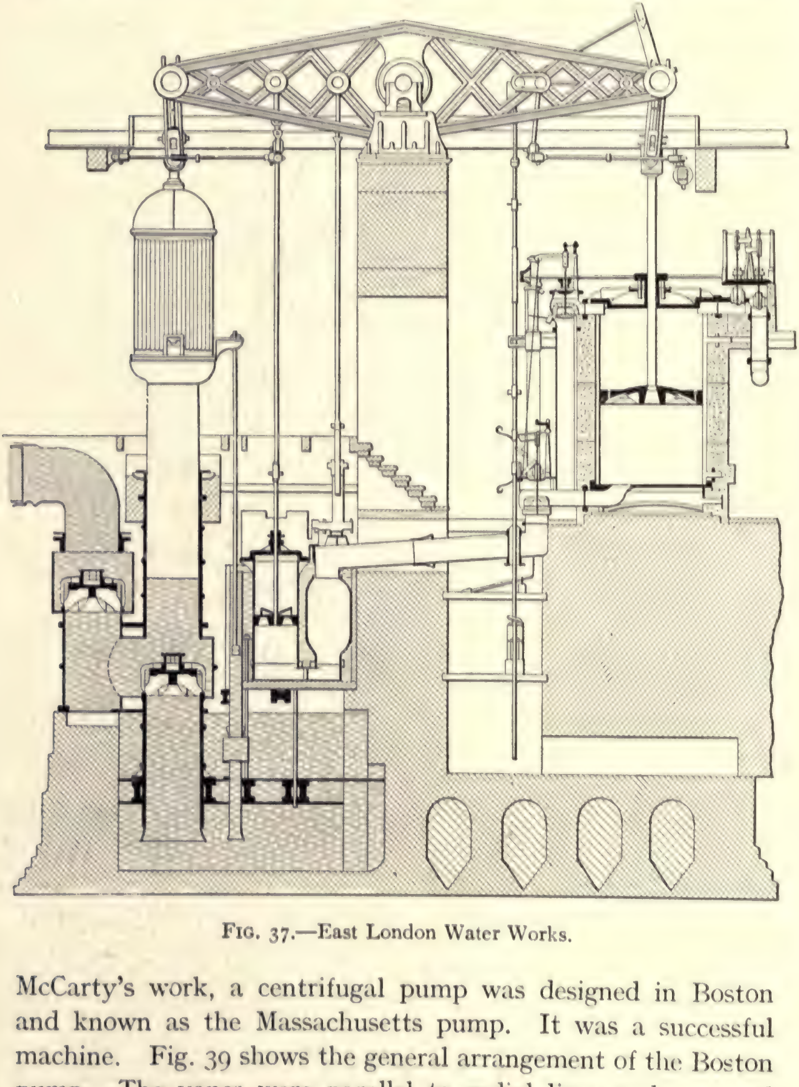


FIG. 37.—East London Water Works.

McCarty's work, a centrifugal pump was designed in Boston and known as the Massachusetts pump. It was a successful machine. Fig. 39 shows the general arrangement of the Boston pump. The vanes were parallel to radial lines and removed

several inches from them. They were placed on each side of a disc, and this runner revolved within a casing.

After McCarty the improvement of this form of pump was undertaken by Blake and Andrews in this country in 1831 and 1839, respectively, and by Appold, Thompson, and Gwynne in England. The original blades of the Massachusetts pump were radial, but those of Andrews in 1846 were curved, as shown in Fig. 40. This pump had the vanes held between two discs.

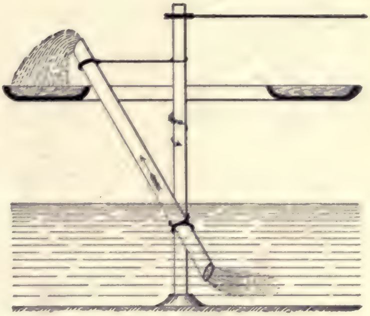


Fig. 38.—Demour's Centrifugal Pump.

This patent was bought by John Gwynne for England, and his firm began the manufacture of these pumps. The development of the centrifugal pump in England is closely connected with

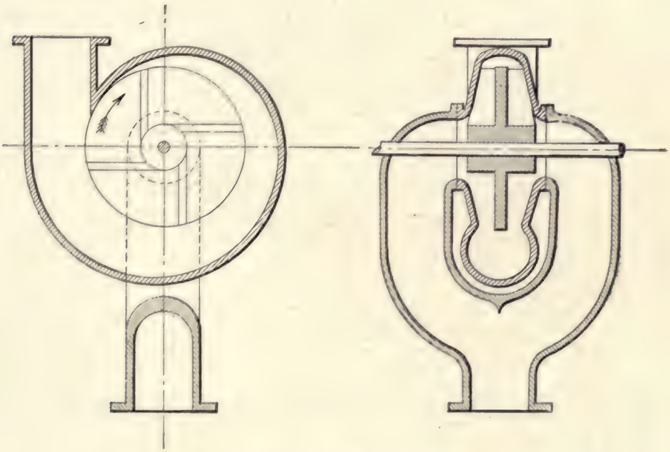


Fig. 39.—The Boston Pump.

the history of this firm. They were the constructors of the most notable installations for many years.

In 1848 Lloyd took out a patent for a centrifugal fan, and Appold began the manufacture of it and applied it to the

lifting of water. In 1851 he exhibited this and showed its practicability. The tests of the pump with the curved vanes showed it to be about three times as efficient as that with straight arms. The advantage of this pump is its ability to lift large quantities of water considering the space occupied by the machine, and its ability to pump small solid particles without clogging. It was originally thought that this was only

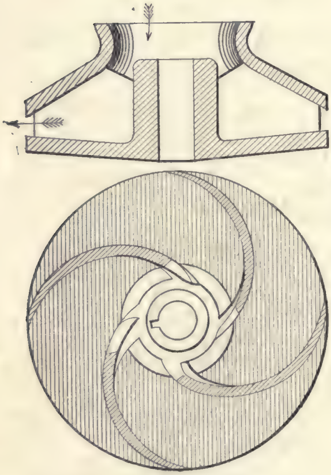


Fig. 40.—Andrews Pump.

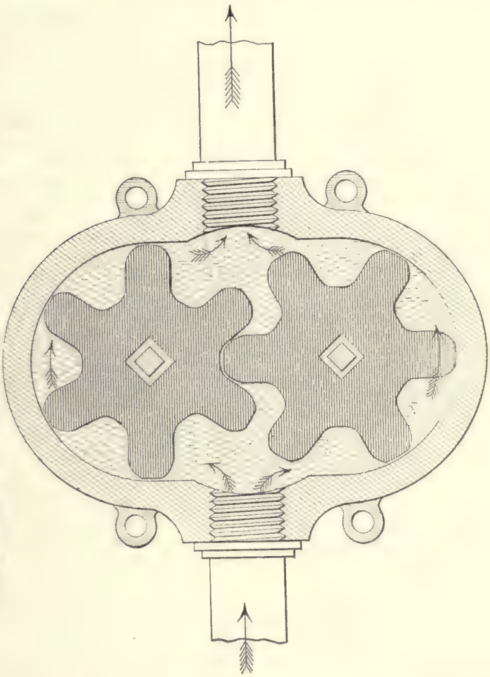


Fig. 41.—Servière's Rotary Pump.

applicable to low lifts, but to-day pumps of this form are used for lifts of several hundred feet.

Another old form of pump in which rotary motion of the parts is utilized is shown in Fig. 41. This is the **rotary pump,** and is an old invention found in the form of Fig. 41 among a collection of models made by Servière, a Frenchman, born in 1593. This is one of the best forms of this type, as will be seen later, when the rotary pump will be examined in detail.

Since the action of this pump is positive, it may be used against

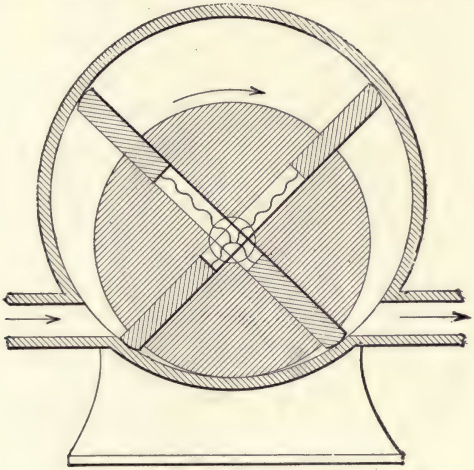


FIG. 42.—Ramelli's Rotary Pump.

high heads, although leakage may be excessive, due to the wear which occurs in the parts. Some claim that this was the invention of Pappenheim, a German, who lived in the seventeenth century.

Ramelli, whose publication of 1588 illustrates a rotary pump shown previously in Fig. 27, uses a slightly different form from that of Servière. In his pump, Fig. 42, a series of flat pistons are driven by a rotating cylinder which is placed eccentrically within the outer casing. These flat plates are held out by springs, as shown in the figure, and the rotation of the inner cylinder forces the water through the machine.

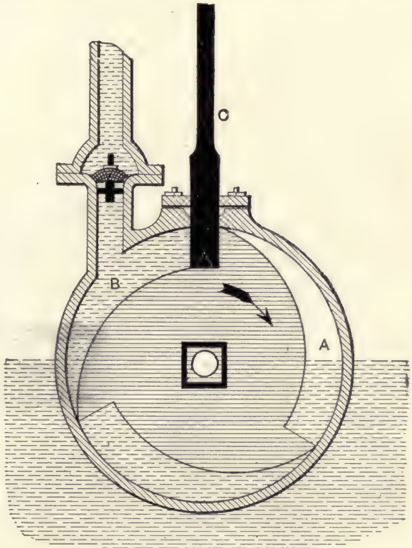


FIG. 43.—Sixteenth Century Rotary Pumps.

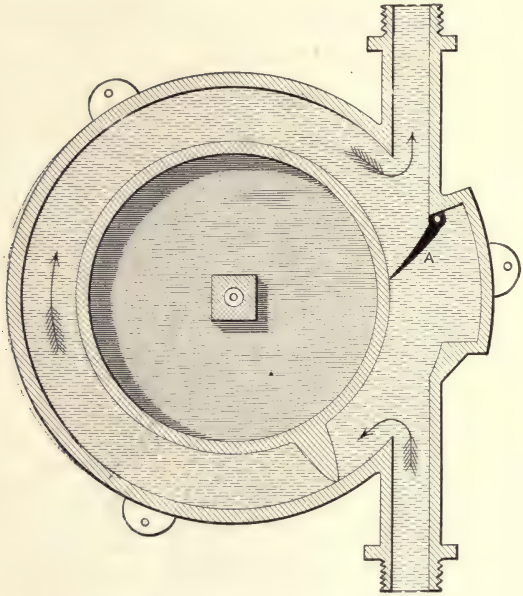


FIG. 44.—Watt's Rotary Pump.

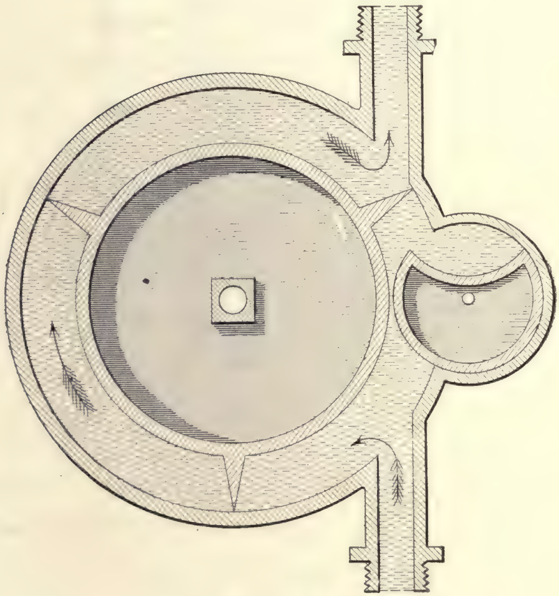


FIG. 45.—Eve's Pump.

Another form of rotary pump of the sixteenth century is given in Fig. 43. As seen in the figure the space *A* is being filled through an opening below the water level while the space *B*, which is closed by the sliding partition *C*, is being discharged. The sliding partition *C* extends from one side of the casing to the other, and slides through the stuffing box. After rising to the highest point it drops by its weight, which is made sufficient to overcome the friction of the stuffing box. The stuffing box shows the form used at that day. The friction of this machine was very great.

A form of rotative pump similar to that patented by Watt in 1782 as a rotative engine is shown in Fig. 44. The operation is clear from the figure; the flap or butment *A* serves to divide the two sides of the pump and when the projecting piece or piston strikes the butment it swings on its pivot. The movement of this butment is controlled by a heavy spring, or by rods and cams, so that it is held against the water pressure from force main, moving only at the proper time. This scheme was altered in 1825 by J. Eve in that he substituted a revolving cylinder for the pivoted butment and inserted three moving pistons for the one. The small drum was driven by gearing from the main shaft at three times the revolutions of the main shaft, as it was one-third the size of the main drum. This is shown in Fig. 45.

In 1805 John Trotter introduced a different form, Fig. 46, in which a plate was driven in such a manner as to touch two fixed concentric drums, its position being radial. The operation of the machine is evident from the figure. There is some chance for leakage after the piston crosses the discharge pipe and before it crosses the suction pipe, so that it is really necessary to have more than one piston. Fig. 47 illustrates another form of this used for water, although it had been used in 1790 for a steam engine.

From these earlier forms a number of new rotaries were developed which finally became a variation of the older form of Servière, as will be seen in the next chapter.

Another old type to be mentioned is the reciprocating

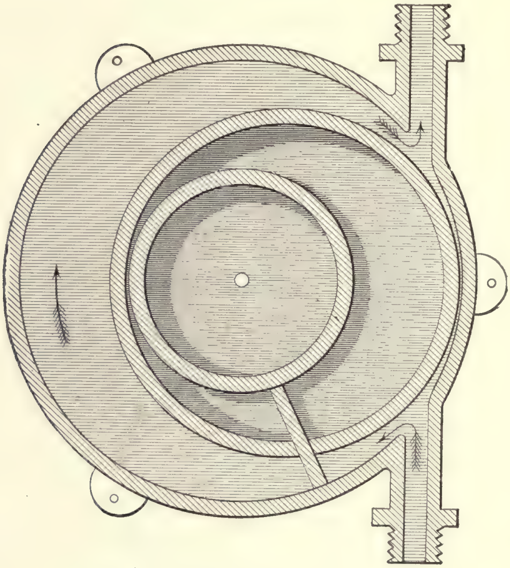


FIG. 46.—Trotter's Rotary Pump.

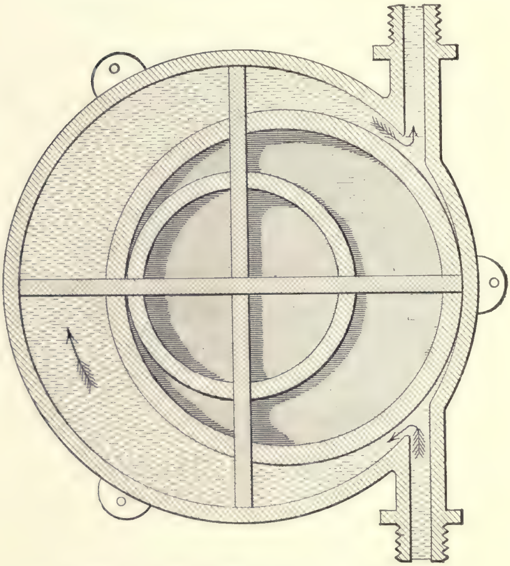


FIG. 47.—Four-Bladed Rotary Pump.

rotary form, Fig. 48. The figure shows the operation of the pump, and a further description is unnecessary. The objection to these and the rotary pumps is the fact that it is very difficult to keep their pistons tight. The rotary pumps have the great advantage that the flow of water is always in the same direction through the pump.

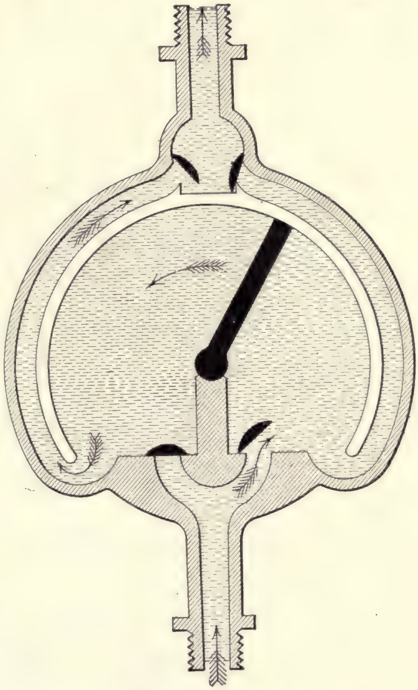


FIG. 48.—Reciprocating Pump.

A pump somewhat allied to the rotary is the **screw pump,** Fig. 49. The pump illustrated was the invention of Révillion, and was patented in Paris in 1830. It consisted of a right- and a left-handed screw meshing together, being driven in opposite directions at the proper speed by means of gears *AB*. The point of one screw touches the root of the other, and thus incloses a definite volume of water between the screw and the walls of the pump chamber, which travels upward as the screw rotates. At the end of the travel this water is forced out at the center.

In passing, it is well to note that the use of pumps for the extinguishing of fires had been common from the earliest times, Fig. 19 being one described by Hero. Until about 1840, however, these were all driven by hand power. Fig. 50 shows a French engine of 1829. This was hauled to the fire by the fire company, which in America was a very important social organization during the first half of the urban history of the last century.

The first steam-driven fire engine of note in this country

was one planned by Captain John Ericsson, in a competition for a prize offered by the Mechanics Institute of New York,

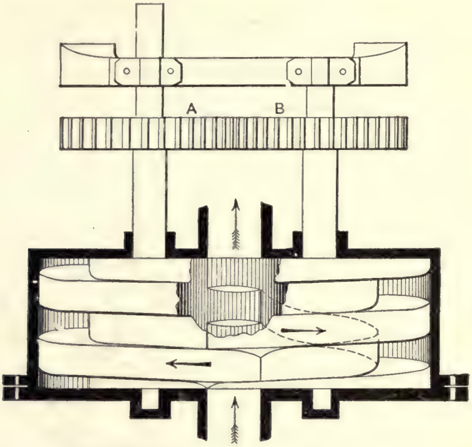


FIG. 49.—Screw Pump of Révillion.

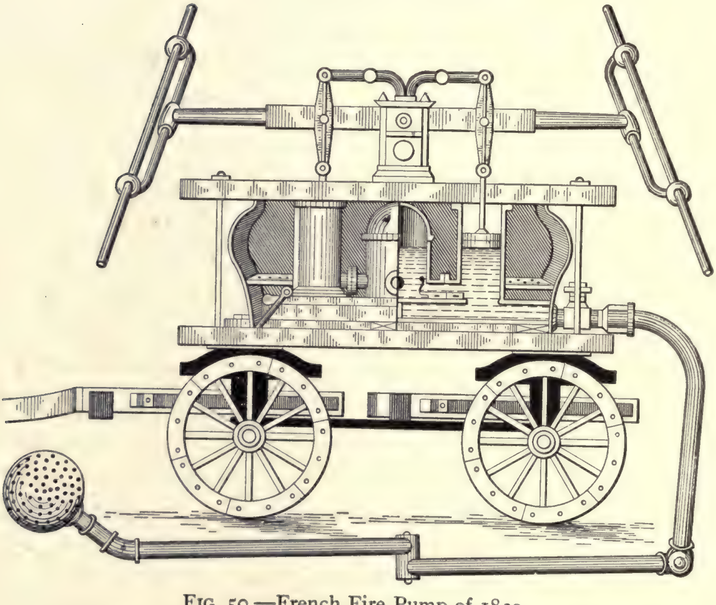


FIG. 50.—French Fire Pump of 1829.

in 1840, although in 1830 Braithwaite and Ericsson brought out a steam fire engine in London. The pump developed

6 H.P., and pumped 150 gallons per minute a distance of 80 or 90 feet. It was drawn by horses and practically was the same as that designed by Ericsson for New York. Before this time stationary steam fire pumps were used.

Fig. 51 gives a view of the Ericsson engine. The boiler was of the locomotive type, the barrel *A* being connected to

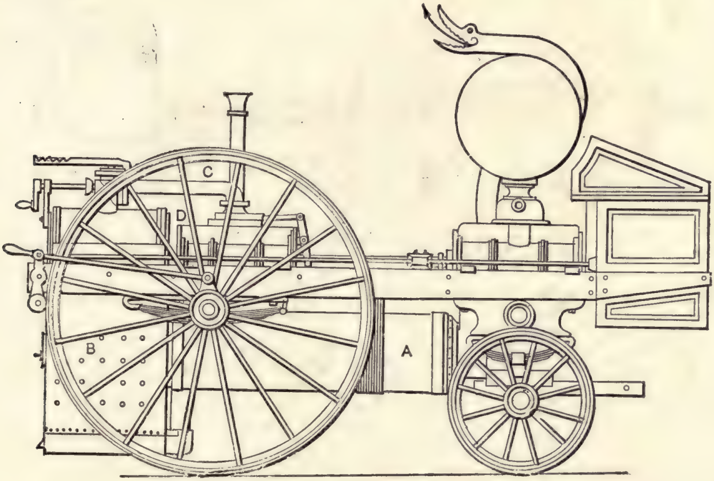


FIG. 51.—Steam Fire Pump of 1840.

the firebox *B*. The steam pipe *C* supplied the cylinder *D* with steam. The water cylinder was in line with the steam cylinder and above it was placed an air chamber. The smoke pipe from the boiler was carried around the air cylinder in the form of a serpent. On the front of the engine was a blowing box which could be worked by the cross-head of the steam engine, or by hand or by a crank attached to the wheels of the engine. The latter arrangement served to force the fire within the firebox when the engine was on its way to a conflagration. This is a forerunner of the modern fire engine, and it is noteworthy that the design was most thoughtfully and carefully made.

The first record of the **hydraulic ram** was that of Mr. Whitehurst of Derby, England. In 1772 he erected a machine shown in Fig. 52. *A* was a spring or reservoir supplying the cock *C*

through the pipe *B*, which was about 600 feet long and $1\frac{1}{2}$ inches in diameter. The cock was 16 feet below the level in *A*, and on closing this after drawing water the momentum of this long column of water was sufficient to force the water

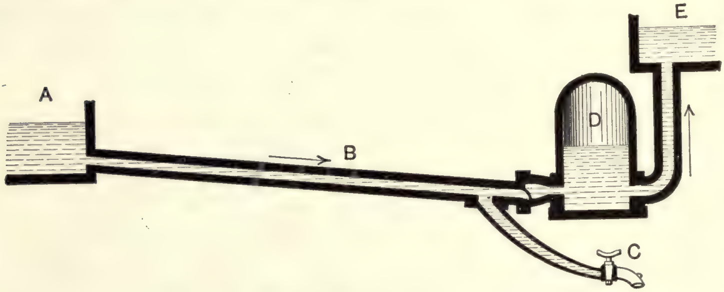


FIG. 52.—Hydraulic Ram of Whitehurst.

into air chamber *D*, which was under the pressure of the higher reservoir *E*.

It was Montgolfier in 1796 who independently invented the same scheme, but made it of more value by using a device

FIG. 53.—Ram of Montgolfier.

which worked, continuously and automatically, in place of the cock *C*. His scheme is shown in Fig. 53. The water descends from the source of supply through *A*, escaping at *B*. When the water has acquired a certain velocity it raises the ball and closes the opening at *B*. The momentum of the water causes an increase of pressure, and this is finally sufficient to open the

valve in C against the high pressure of the discharge. The valve at B may be of the disc form, but opening downward; the principle, however, is the same in all cases.

The development of the ram in the years which follow the work of Montgolfier consists in improvement in details, the

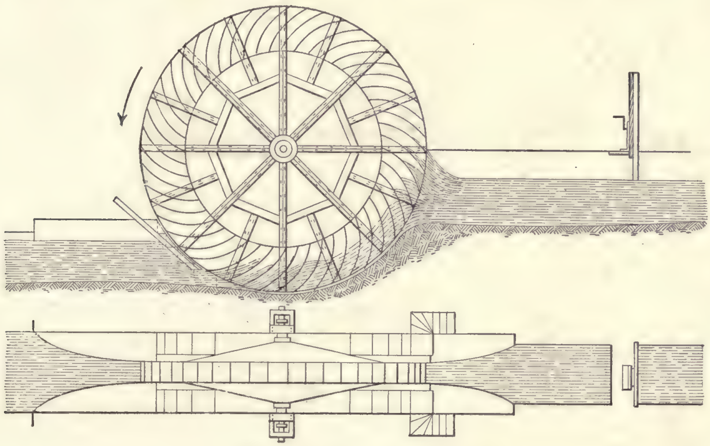


FIG. 54.—Flash Wheels.

latest forms of the present day being quite similar to this early type.

Scoop wheels or **flash wheels,** Fig. 54, were used from early times. They were in reality water wheels turning backward. These, as will be seen, have been used to some advantage in later times. They were used extensively in Holland about the time of the introduction of the steam engine.

CHAPTER II

RECENT HISTORY

THE year 1840 marks an era in the history of pumping machinery, for it was in that year that Henry R. Worthington began his brilliant inventions, which have led to most of the modern forms of steam pumps.

Before this time the boiler-feed pump was usually driven by the main engine through some extension from the piston or pump rod or by an auxiliary rod from the walking beam. Worthington was concerned in the design of a steamboat for canal navigation. It happened that when the boat was stopped at locks or by obstructions the attendants had to resort to the use of hand pumps to keep the boiler properly filled. An independent steam pump was thought necessary, and on September 7, 1841, the pump shown in Fig. 55 was patented.

In the illustration this pump is mounted on a base for exhibition purposes, although it was originally bolted to the side of the boiler setting at the front. Steam enters the cylinder *A* through the pipe *B*, the steam being directed to either end by a valve in the steam chest *C*. In the position shown, the valve has just been moved so as to admit steam to the right-hand end of the cylinder. This drives the piston, piston rod, and plunger to the left. The plunger *D* drives the water from the cylinder *G* through an ordinary conical valve in the valve box *E* to the feed pipe *F* and from there to the boiler. As the rod moves to the left the arm *H*, attached to the rod, moves the tappet rod *I* to the left. Finally near the end of the stroke the right tappet *J* strikes the lever *K*, pivoted at its center, and forces it against the sloping top of the rod *M*. This forces the rod *M* down against the pressure of a spring. After the lever *K* passes over the point, the spring pressure suddenly

forces the lever K over to the left away from the tappet. This moves the arm N to the right, which controls the steam valve through the axis of the arm N so as to admit steam to the left-hand end of the cylinder A.

The piston, piston rod, and plunger now move to the right,

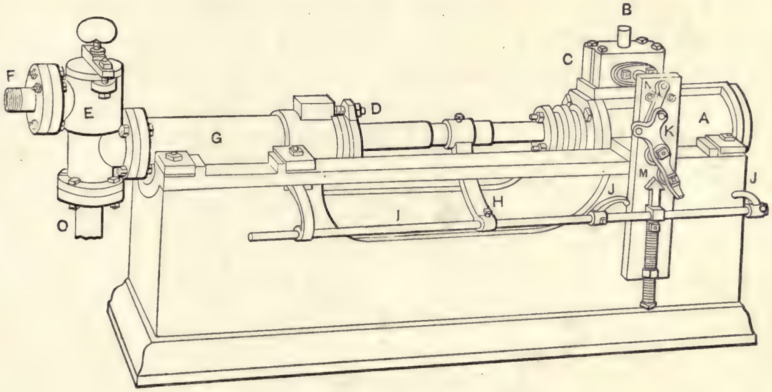


FIG. 55.—Original Worthington Pump.

and water is sucked into the cylinder G through a conical valve at the bottom of E and the suction pipe O. The left-hand tappet drives the lever K to the right and finally the spring forces it over, suddenly reversing the motion. The spring action was necessary to properly reverse the steam valve, as a tappet

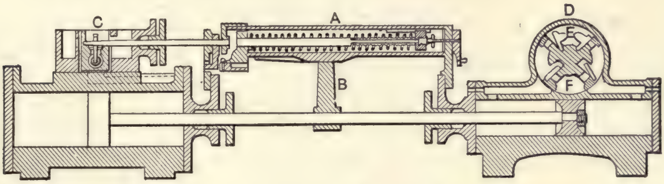


FIG. 56.—Spring-Thrown Valve.

alone would permit the valve to cut off steam from each end when the motion was slow. It is necessary to have the valve reversed positively while the pump has positive motion, however slow.

The admission of steam to this particular pump was controlled by a float in the boiler so that when the water became

low the pump would start automatically and continue until the float would again cut off the supply of steam. The pump shown in the figure was in service for twenty-five years and was finally bought back by the Worthingtons as a cherished relic.

In 1844 Worthington used a helical spring in a casing *A*, Fig. 56. The arm *B* pressed against a helical feather or projection on the casing, turning the casing against the action of the spring. When the arm went beyond the end of the feather the spring forced the casing over suddenly, thus moving the valve at *C* by turning the valve rod, moving the valve across the cylinder, perpendicular to the piston motion.

The water valves at *D*, in which *E* is the discharge space and *F* the suction, are of the forms used in many pumps of

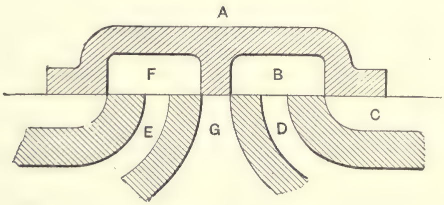


FIG. 57.—*B* Valve.

that day. The pistons, stuffing boxes and cylindrical parts are quite similar to those of the present.

The *B* valve, Fig. 57, was invented for the purpose of admitting steam to the right-hand end of the cylinder by a motion to the right when steam is above the valve. In the figure it is seen that when the valve is moved to the right, the steam above the valve at *A* will enter the space *B* through the space *C* and thus pass to the right-hand end of the cylinder through *D*. At the same time the left-hand end *E* is connected to the exhaust port *G* through the cavity *F*. This is necessary when a slide valve is moved directly by the motion of the piston, since the motion to the right moves the valve to the right and with this movement steam is admitted to the right-hand end, reversing the pump.

The next improvement was a steam-thrown valve, Fig. 58. This was in 1849. In this arrangement an auxiliary valve rod not shown moved an auxiliary valve, admitting steam to the right or left of the piston *A*, which forced the small cylinder to the right or left, thus moving the main valve, which was

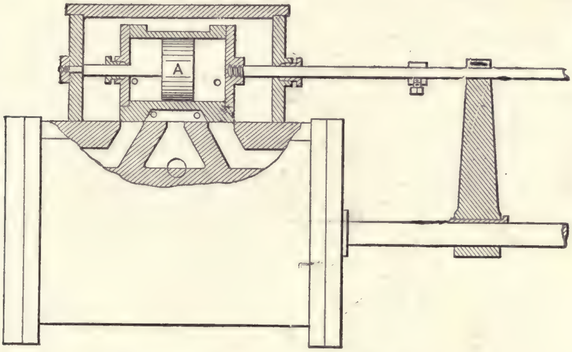


Fig. 58.—Steam-Thrown Valve.

part of the small cylinder. The excessive friction from the steam on top of the auxiliary piston led to the design of a balanced valve which took steam in through the ordinary exhaust passage and used the so-called steam chest as an exhaust chest.

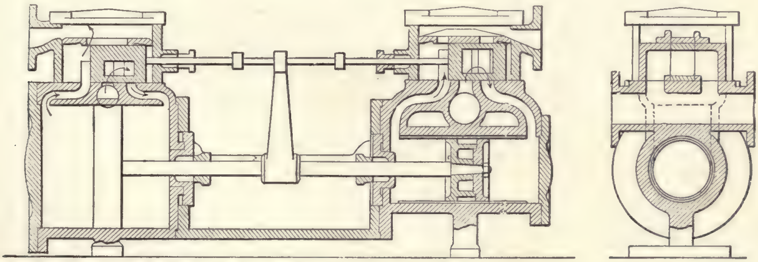


Fig. 59.—Positive Water and Steam Valve.

Fig. 59 shows a pump built about 1850 in which both the water and steam valves were controlled by the movement of the valve rod. This was in a measure a water dash pot for the purpose of stopping the motion of the pistons. Worthington used this in some cases, but he did not advocate its general use.

In 1849 Worthington, with his partner, Wm. H. Baker, patented a relief-valve motion. This scheme is shown in Fig. 60, where radial water valves or clack valves are used. The invention consisted of the use of two water passages *AA* at each end of the water cylinder, so that when the water piston uncovered the inner of these ports the pressure in front of the piston was relieved suddenly and the steam in the steam cylinder drove the piston to the extreme end, moving the main slide valve *B* over by means of the arm *C* and the tappets on the valve rod *D*. This same scheme was applied by cutting grooves in the water-cylinder bore at each end.

The aim in all of these later pumps was to simplify the

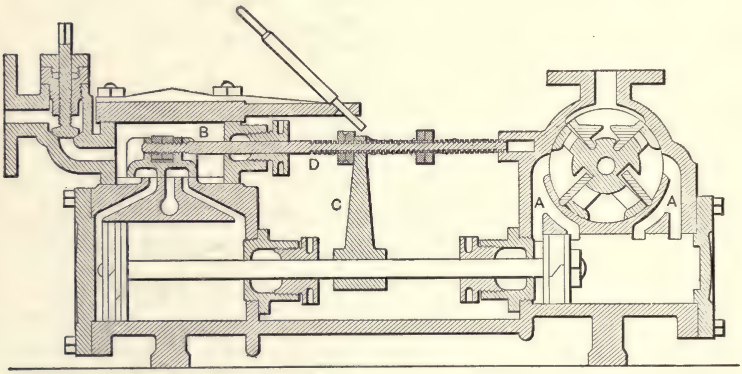


FIG. 60.—Relief Valve Motion.

mechanism so that the pump would do the work in a positive manner and still be simple to care for and to operate.

In 1850 Worthington made another improvement by substituting a number of small valves for the four large valves used on pumps heretofore, and also employed a plunger and ring in place of the piston of former pumps. The pump, Fig. 61, was used on the steamer "Washington." There were thirty-six of these small valves arranged on valve decks as indicated in the figure. The large number of small valves gave the requisite amount of opening with a small lift and hence the amount of leakage passing the valves when the pump was

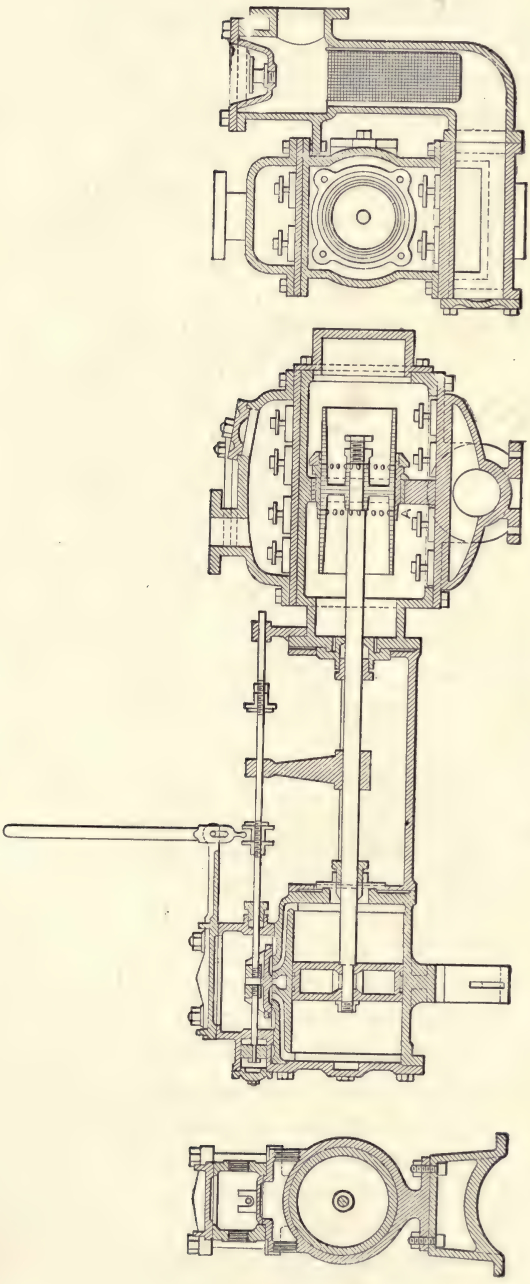


Fig. 61.—Pump for the S. S. Washington.

reversed was greatly reduced. The valves were of rubber, one-half inch thick.

The arrangement of the plunger and ring of Fig. 61 is markedly different from the pistons of the former figures, and Worthington adopted it for several reasons:

1st. It gives ample space above and below for the valves.

2d. The ample space around the plunger forms a subsiding chamber where harmful materials may settle out of the way of the plunger packing.

3d. The constant protrusion of the plunger tends to carry foreign matter away from the packed joint.

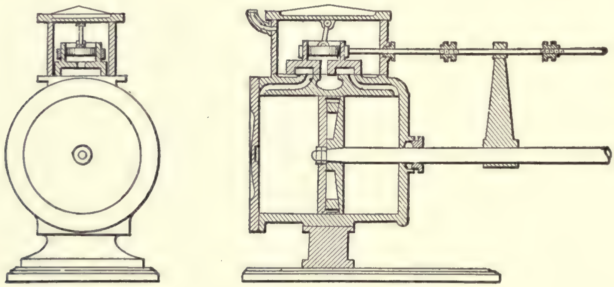


FIG. 62.—Steam End of Savannah Pump.

4th. The construction makes the removal or renovation of the parts an easy matter.

5th. The deflection of the water currents is less than in ordinary arrangements. The friction is very small.

It is to be noted in passing that this design simplified the construction of the water end of the pump as far as the foundry-work and the machine-shop work were concerned. The use of a solid metal packing around the plunger as seen at *A* was an innovation, but it has proved a success. It was made long and with clear water the wear was not sufficient to cause excessive leakage for some time. A steam-thrown valve was used in this case. Worthington was not satisfied with this method of valve operation and after his invention of the duplex pump in 1859 he rarely used it, although inventors such as Knowles, Blake, and others continued to use it.

In 1854 Worthington erected his first water-works pump for the city of Savannah. It was a compound pump of peculiar design, in that the high-pressure cylinder was at the center of the annular low-pressure cylinder. Another new feature of the pump was the balancing of the steam valve by carrying part of the pressure on the back of the valve by a piston supported from the steam-chest cover as shown in Fig. 62. This figure illustrates the use of this balancing method on an ordinary cylinder. The Savannah engine was a success and a duplicate of it was built for Cambridge, Mass., in 1856. This engine, and a companion engine, built soon after, were of the following dimensions:

High-pressure cylinder	12 inches diameter
Low-pressure cylinder	25 "
Plunger	14 "
Length of stroke	25 inches
Capacity	300,000 gals. per 24 hrs.

This engine was the best of its day, as was shown by tests made for the Brooklyn Water Works in 1857 and 1859, when that city was about to install new pumping engines. The results of these tests as shown in the report of Mr. James P. Kirkwood, chief engineer of the Brooklyn Water Works, were as follows:

Date of Test.	Name of Engine.	Duty in ft. lbs. per lb. of Coal.
April, 1857	Worthington engine, at Cambridge	669,411
June, 1857	" "	675,746
Jan., 1857	Cornish engine, at Jersey City	628,233
July, 1857	Hartford crank engine	646,994
July, 1857	" "	614,426
Jan., 1860	Brooklyn new rotative engine	601,407
June, 1856	Cornish engine, Philadelphia	589,903

As a result of this the Worthington pump was selected and a modification of the Cambridge pump design, known as the duplex pump, was accepted; this was, however, abandoned on account of trouble with the contractor and it was not until 1863 at Charlestown, Mass., that this important type was installed in a water works. It will be remembered that the term duty as used by Moreland and Watt meant the useful

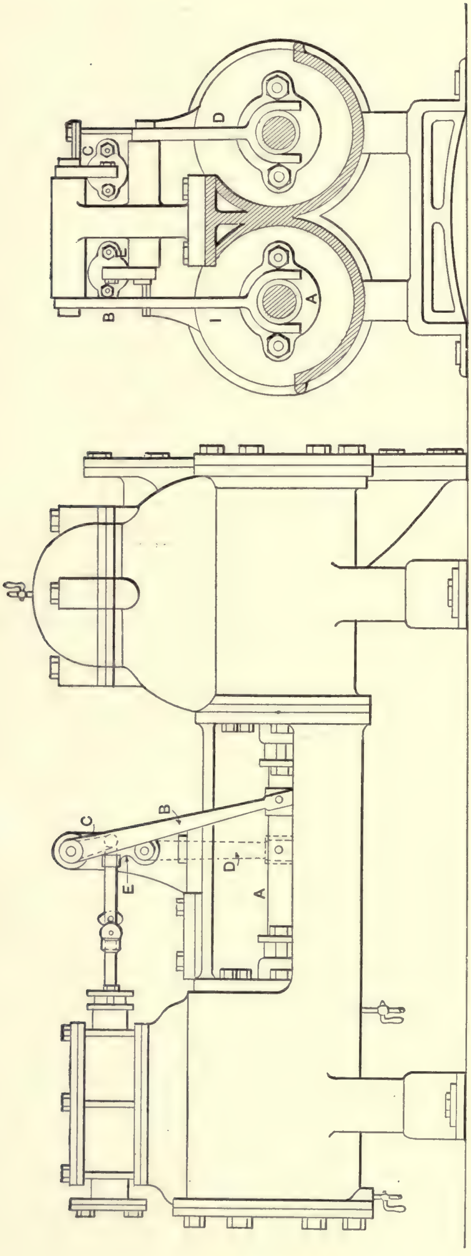


Fig. 63.—Duplex Pump.

work per bushel or hundred weight of coal, but in later years the duty was figured as the useful work per 1000 pounds of steam or per 1,000,000 British thermal units.

The date of 1859 is that of the invention of the duplex pump, one of the simplest contrivances for operating the valves of a pump, doing away with the complex arrangements used heretofore for this work. This then became for many years the standard type of pumping engine, as the Corliss engine became the standard mill engine.

To make clear the operation of the duplex pump a modern form of boiler-feed pump, Fig. 63, is illustrated. This consists really of two pumps placed side by side. In this figure the piston rod *A* of the front pump has just reached the right-hand end of its stroke, and the bell-crank lever *BC*, which extends from the front piston rod to the back valve rod has moved the *D* slide valve of the back pump to the right, uncovering the steam port of the left-hand end of the rear pump, causing that pump to move to the right. This motion is transmitted to the front valve stem through the reverse lever *DE*, which connects the back piston rod to the front valve rod. The *D* slide valve of the front pump is moved to the left, admitting steam to the right-hand side of the cylinder and thus the piston of the front pump moves to the left. This motion, in turn, causes the back pump to move to the left and this then moves the front valve so that the piston moves to the right, when the operation is repeated.

The action of the pump may be represented by the diagram, Fig. 64, in which vertical distance represents time and horizontal distance represents motion of the pump. The solid line represents the front pump and the dotted one the rear pump. At the end of each stroke there is a period of rest while the other pump follows the stroke of the first one. This is accomplished, as will be explained later, by the method of moving the valve by the valve rod or its equivalent.

Not only is the claim for a simpler valve gearing made or this pump, but there should also be a steadier discharge of water, because as one pump nears the end of its stroke the

other one discharges water to keep up the flow while the first reverses.

This duplex pump was one in which, by the use of an additional unit, a simple form of valve gear was obtained. While Worthington was inventing steam-thrown valves of various forms previous to his invention of the duplex pump the same

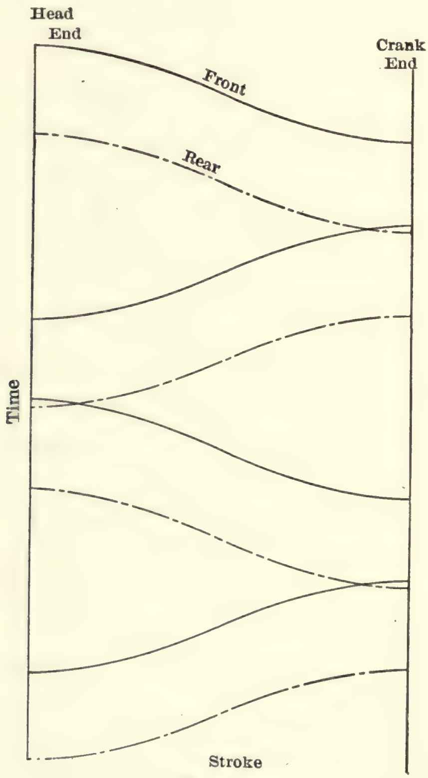


Fig. 64.—Action of Duplex Pump.

problems were being investigated by others in this country as well as abroad. It is not the intention of this book to follow all of the different forms of simplex pumps invented from the time of Worthington to the present, as his inventions mark the era. Later a number of modern simplex mechanisms will be examined in detail to illustrate what had been done and

what forms have survived. It is necessary at this point, however, to mention such names as Blake, Cameron, Knowles, Earle, Cope, Maxwell, Marsh, Silver, Dean, Davison, and Gordon in America; and Moreland, Thompson, Baummans, Tyler, Clarkson, and Davies in Europe, as some of those who had been working on this problem. Their work is of interest in comparing the complicated manner of the earlier forms with the simpler forms of to-day.

The introduction of the duplex pump, at least while the patent lasted, caused considerable argument as to the relative merits of the simplex and duplex forms, the simplex manufacturers claiming a lack of positive length of stroke for the duplex against a full stroke of the simplex to offset the greater complication. The proper adjustment of each pump, however, will give a proper stroke.

Pumps of the duplex type were introduced into water works in 1860, and in 1863 one of 5,000,000 gallons per twenty-four hours was installed by Worthington at Charlestown, Mass., and in 1871 a larger one for 19,000,000 gallons was built for the city of Philadelphia. The earlier engines were single expansion on the steam end.

The pumping engine of Mr. George Shields for the city of Cincinnati, which was built in 1861, was one of the largest of its day; it was to lift 9,600,000 gallons per twenty-four hours against a head of 170 feet. It was direct acting and had no balance beam. The steam cylinder was 100 inches in diameter and had a stroke of 12 feet. The pump cylinder was 45 inches in diameter. This engine was one of the unfortunate structures of American engineering in that it cost the municipality many times its original estimated figure; but to its credit it may be said that twice it was the means of saving the city from a water famine, when the other pumps gave out. The use of vertical engines in which the steam and water cylinders were placed over each other was quite common, as, for example, in the Bull Cornish engines, but when it was desired to eliminate the inertia bob weight to cut down the weight of the engine, fly wheels were introduced. The fly wheels were driven from a

beam in most cases, but in 1868 Richard Moreland, Jr., and David Thompson invented the direct-acting engine shown in Fig. 65. In this the cross-head was connected to the pump plunger by two or four rods which spanned the crank shaft. The cylinder was supported by A frames. The pump barrel was bolted to the base of the engine. The pump was single acting and the general arrangement was excellent and simple.

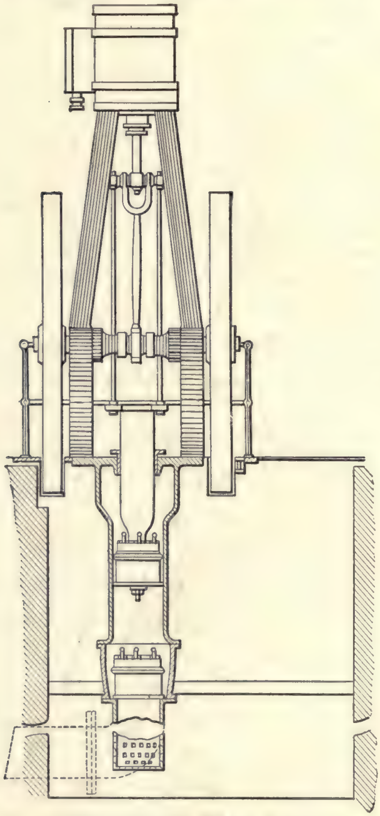


FIG. 65.—Fly Wheel Pump.

This type of engine was installed in 1868 and 1876 at the Eastbourne Water Works in England. It proved to be so successful in operation and needed so little repairing after being in service until 1881 that new engines for that plant were made of the same type.

The general arrangement of steam end and pump for this machine will be seen to be quite similar to the modern American pumps. The design was good and one of the simplest for the application of the fly wheel to the pump.

About this time a unique pump was introduced by Birdsill Holly to care for his direct-pressure system, which dispensed with a reservoir or standpipe, obtaining pressure direct from the engine. He introduced the system in 1866 in Lockport, N. Y., using a pump driven by a water wheel, but in 1871 he used his quadruplex engine at Dunkirk, N. Y. The four pumps, Fig. 66, were placed in tandem with the steam cylinders, which were arranged in pairs, each acting on a crank. The two

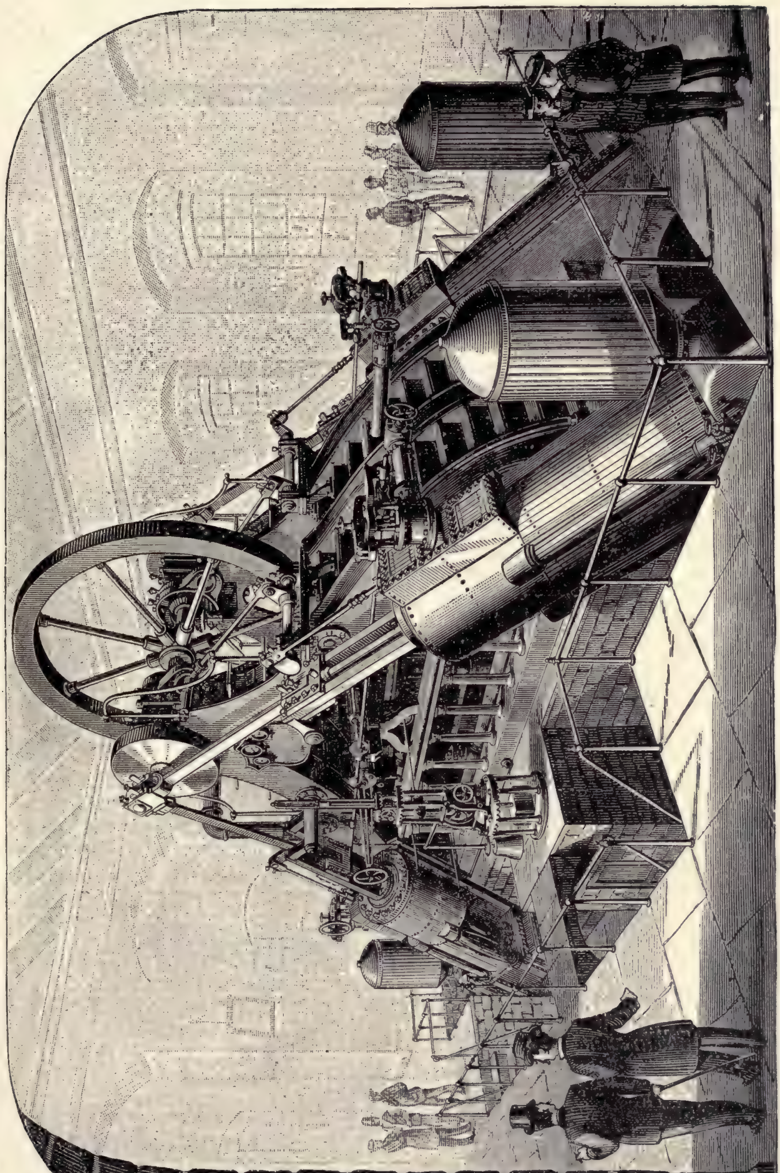


FIG. 66.—Quadruplex Pump.

cranks were arranged at 135° and the frame of the engine was such that the center lines of the engines intersected at 90°. The figure shows the construction of the engine.

Such an engine was necessary for this system, as the engine had to start from any position as soon as the drop in pressure in the mains moved the governor. This system saved the expense of a reservoir, but the necessity of keeping the engine under steam continually made the steam use and the labor

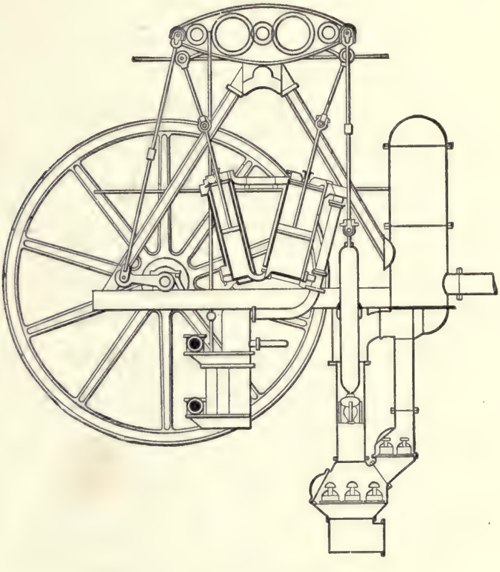


FIG. 67.—Levitt's Lynn Pump.

expense very high. The engine could be run with any number of steam or water cylinders, and the engines could be used as single-expansion cylinders or by exhausting from one to the other three, as was done in 1874 at Rochester, the advantages of compounding could be had. On account of the intermittent action of these engines their duty was not better than that of other pumps. A duty of 86,176,315 foot-pounds was obtained on the 6,000,000 gallon pump at Buffalo, N. Y., in 1879.

The first important compound steam end for water works in America was designed in 1872 by Frederick Graff, chief

engineer of the water department of the city of Philadelphia. Compound pumps were built by Simpson & Co. of England, in 1848, although most of the pumping engines were of single-cylinder type.

This Graff engine was followed in 1873 by an engine designed by Mr. E. D. Leavitt, jr., which was quite similar to the Graff engine in its theoretical operation, although different in its arrangement. It was intended for the water works of Lynn, Mass.

As shown in Fig. 67, this had two inclined steam cylinders and a pump which was arranged to discharge on each stroke, although it was single acting on the suction side. This was accomplished by the plunger attached above the pump bucket. The supplementary pipe and valves connecting the two ends of the pump cylinder were for the purpose of reducing the friction. The valves were of large diameter and double beat, that is, each valve had two discharging edges. This type of pump end was first built in 1848 and was known as a Thames-Ditton pump. The figure also shows the application of the fly wheel to the pump for the purpose of using steam expansively. The cylinders were steam jacketed and the steam valves were of the gridiron type, driven by cams. The double-acting air pump was driven from the beam.

This pump gave a duty of almost 104,000,000 foot-pounds per 100 pounds of coal on a 52-hour test, when pumping about 5,000,000 gallons per day. The duty for its year's record was about 75,000,000. The dimensions of the pump are given below:

Diameter of high-pressure cylinder	17½ ins.
" low-pressure cylinder	36 "
" high-pressure piston rods	3 "
" low-pressure piston rods	3¾ "
" air pump	11¼ "
" pump barrel	26.1 ins.
" plunger	18½ "
Length of stroke of steam and water pistons	7 ft.
" " air pump	44½ ins.
Distances between end centers of the beam	11 ft.
Weight of fly wheel	10.7 tons
" beams	4.2 "
" moving parts connected with beams	5 "
Length from pump to top of vertical pipe reservoir	1904 ft.
Height of top of vertical pipe above bottom of well	163.34 ft.

This pump with its walking beam may be taken as an example of many of the pumping engines to be found in Europe and America, although the arrangement of the cylinders in an inclined position was novel. Practically all of these machines, with the exception of Graff's for the Philadelphia Water Works,

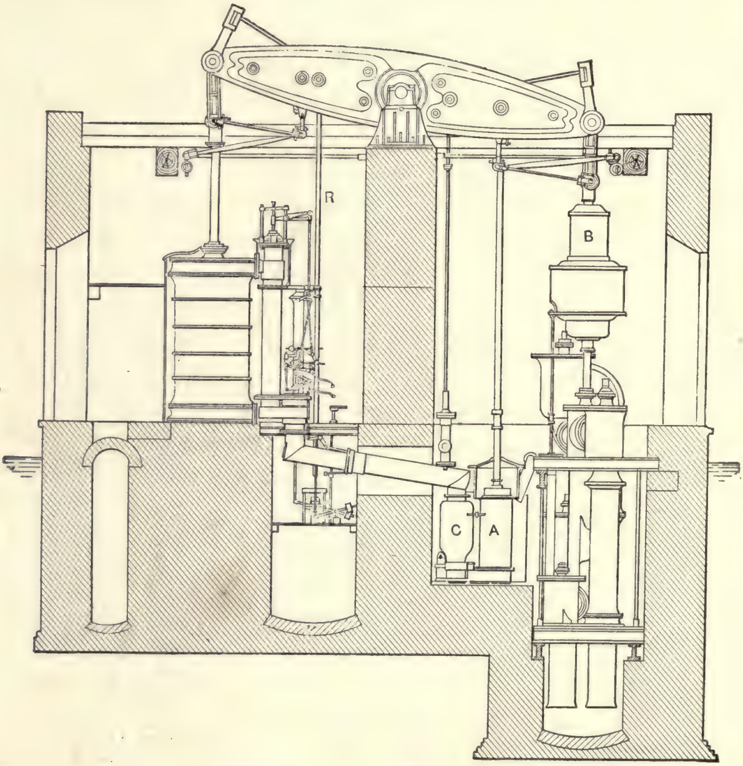


FIG. 68.—English Pump of 1866.

were with single cylinders, although compound engines had been known. To give an idea of one of the English pumps of this period the West Middlesex pump of 1866 is shown in Fig. 68, as this illustrates the general arrangement of such pumps. The steam cylinder was 80×120 inches, while the water cylinder was $24\frac{1}{8} \times 120$ inches. The pump was double acting, although the steam cylinder was single acting. The large acorn-shaped box

B on the pump rod directly beneath the parallel motion is the balance box, which was loaded so as to force the water from the pump against a head of 200 feet on the down stroke. During this stroke the two ends of the steam cylinder were connected through a 15-inch equilibrium pipe as in the old type of Watt engine. The weighted box pulled the piston up until the piston reached the top of the 10-foot stroke, when the plug rod *R* reversed the valves, bringing live steam on top of the piston and connecting the lower end of the cylinder to the condenser *C* through a 19-inch exhaust valve. The steam on the upper side then drove the piston downward, the air pump *A* driven from the main beam maintaining the vacuum in the condenser. This pump was provided with a safety device so that pressure would be held in the discharge chamber in case the discharge pipe should break. This pump is the equivalent of Watt's original pump, although it was worked with steam at 40 pounds pressure. The water pressure was 200 feet and the pump made $16\frac{1}{2}$ strokes a minute.

Several important inventions of Worthington's should be mentioned here: the dash relief valve, the rotary pump valve, and the cross connection for compound pumps.

The dash relief valve was used on pumps to regulate the length of the stroke and prevent pounding. The motion of the piston could be stopped by cutting off the exhaust steam before the end of the stroke and compressing the steam in the space behind the pistons. This was done by having the piston ride beyond the port to the exhaust passage, but in order to introduce live steam into the cylinder in this cushion space it was necessary to use another passage beyond the one for the exhaust. This made five steam passages in the cylinder, Fig. 69, the two outer for steam and the three inner ones for exhaust. It is seen from the figure that when the piston reaches the position shown, the steam to the left of the piston, retained behind the piston after passing *B*, has been compressed into the clearance space and into the passage *A*. This acts as a cushion, and as soon as the valve moves to the right high-pressure steam enters and drives this piston to the right. If the piston is

brought to rest too far from the end of its stroke, a valve *D* is opened slightly between the passages *A* and *B* and some of the steam compressed in *A* exhausts into *B* and then to *C*,

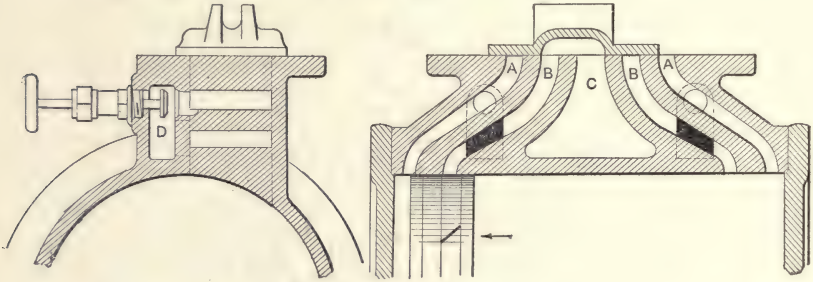


FIG. 69.—Dash Relief Valve.

thus allowing the piston to travel farther. If *D* is opened too much the piston will strike the cylinder head, causing pounding, and it is then necessary to close the valve *D* slightly. The

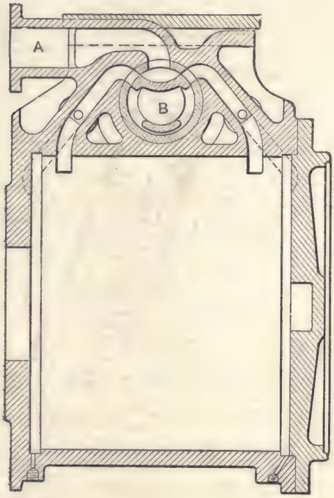


FIG. 70.—Rotary Steam Valve.

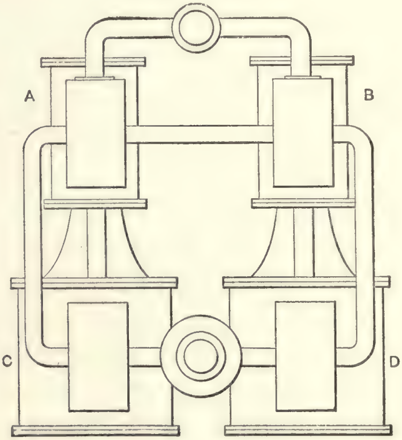


FIG. 71.—Cross Connections.

application of the rotating valve was the application of the common valve used by Corliss to the steam cylinder of the pump, Fig. 70. This valve operates in the same manner as a slide

FIG. 72.—Worthington Water Works Pump of 1873.

valve, the steam or exhaust being conducted by the passage *A* or the center of the valve casting *B*. The dotted passage leading to the end controlled by a dash relief valve acts as the

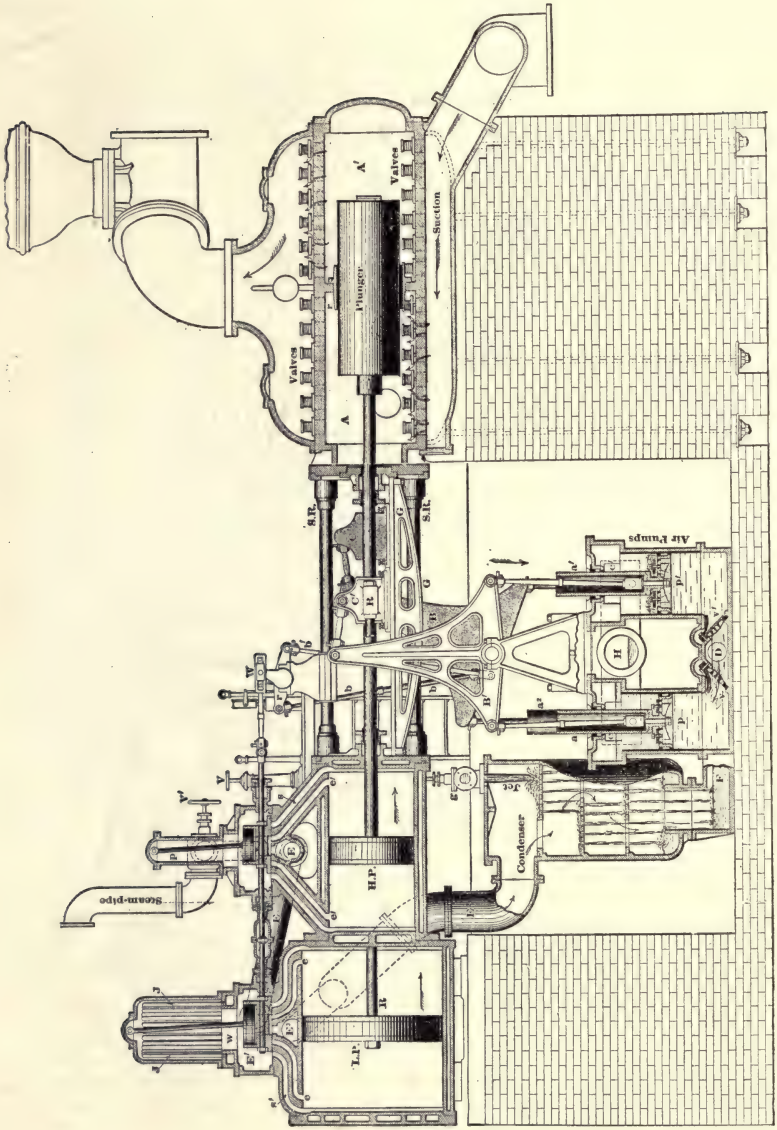


FIG. 73.—Worthington Compound Water Works Pump.

passage for the starting steam. In such an arrangement the piston starts slowly because all of the steam to start it would have to pass the relief valve.

The cross connection of the low-pressure cylinders, Fig. 71, is one which is used to supply steam for the demands of either cylinders *C* or *D* from the exhaust of both of the high-pressure cylinders *A* or *B*. Without the cross connection, the steam from *A* would be used only in *C* and that of *B* in *D*, but by this cross connection of the valve chests a steady motion is obtained.

The duplex Worthington pump was also made in the compound form, and one tested in 1873 at the Belmont Water Works of Philadelphia gave a duty of 54,416,694 foot-pounds on 100 pounds of coal. This pump had steam cylinders of 29 and 50¼ inches in diameter, while the water cylinder was 22½ inches in diameter. The common stroke was 50 inches. The capacity of the pump was 5,000,000. Fig. 72 shows the general form of this pump. A similar one at Newark, N. J., gave 77,358,478 foot-pounds duty.

The difference between the duty of the Leavitt pumping engine and that of the direct-acting duplex Worthington is rather marked, and where the engine is to run at full capacity for considerable length of time these figures show conclusively that such a high-duty engine, even though its cost is much greater than the simpler pump, would prove to be a paying investment. However this may be, when the use of the pump is quite intermittent, if the pump is too large for the average consumption of water, the loss in starting up, together with interest, depreciation, insurance, taxes, and repairs on the more expensive machine might make the cost of pumping greater than it would be with the less efficient machine.

This important fact was emphasized by Mr. Worthington and undoubtedly accounts for the greater use of these duplex machines in the years which immediately followed.

At the Centennial Exhibition of 1876 the Worthington pump was used to supply all of the water. The pump is shown in Figs. 72 and 73. The dimensions were as follows:

Diameter high-pressure cylinder	29	ins.
" low-pressure cylinder	50¼	"
" water plunger	22¼	"
Stroke	48	"
Air-pump diameter	29¾	"
" stroke	24	"

From the sectional view of the pump it is seen that the steam and water cylinders are in tandem on each side of the pump and that the valves are of the swinging-piston balanced type. The cylinders have separate steam and exhaust passages and the outside cut shows the adjusting relief valves. The cylinders were jacketed on the barrel and heads. It will be noted that the valves are driven from bell-crank levers attached to the walking beams of the air pumps by connecting rods. The air-pump beams are worked from cross heads on the piston rods through short connecting rods. It is evident that the connections of the rods, beams, and levers are such that the motion of one engine controls the valve of the other engine.

The construction of the jet condenser and the air pump is seen in the picture. The water end of the pump shows the small valves advocated by Worthington as well as the use of the plunger and ring. The direct path of the water through the pump is one which diminishes the friction loss in the machine.

The centennial year 1876 marks the installation of another Leavitt pump at Lawrence, Mass. This pump was built on the same lines as the Lynn pump, but it gave a duty of 117,550,800 foot-pounds per 100 pounds of coal, making a new record.

The Corliss engine of the Centennial Exposition was one of the most remarkable features of the exhibition. In 1878 Mr. George H. Corliss used his engine in the construction of a pump for Pawtucket, R. I. This pump, Fig. 74, consisted of two steam cylinders, each one in tandem with a water cylinder. The tail rod from the water end was attached by a connecting rod to a vibrating lever, pivoted in the base and joined by a rod to the crank of the engine. By use of the fly wheel in this and the other fly-wheel engines, steam could be used expansively without the use of heavy reciprocating parts. The fly wheel

was mounted on the shaft carried by the bearings, which were supported by the air chambers of the pump. A diagonal brace was carried from the bearings to the main center pedestals.

The steam cylinders were 15 and $30\frac{1}{8}$ inches in diameter, 30 inches in stroke, and were furnished with the Corliss valve gear. They were steam jacketed on the barrels, heads, and

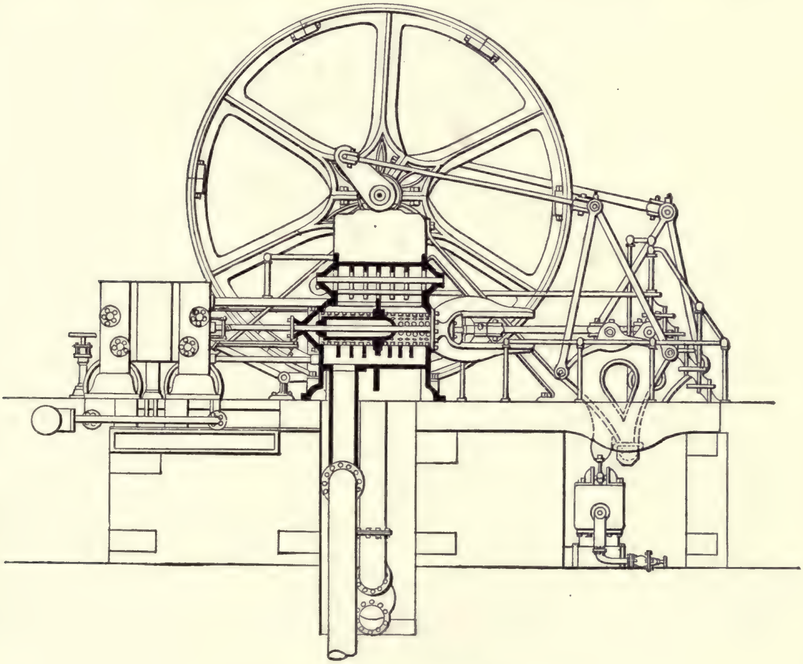


FIG. 74.—Corliss Pump for Pawtucket.

valve boxes. There was a receiver between the cylinders, the volume of which was equal to that of the low-pressure cylinder. The drips from the jackets were delivered into the boiler feed, heating it, while the drip from the receiver was passed through a coil in the boiler flue and returned to the receiver in a superheated condition. The steam throttle valves were so connected with the governor that they limited the speed to 52 revolutions.

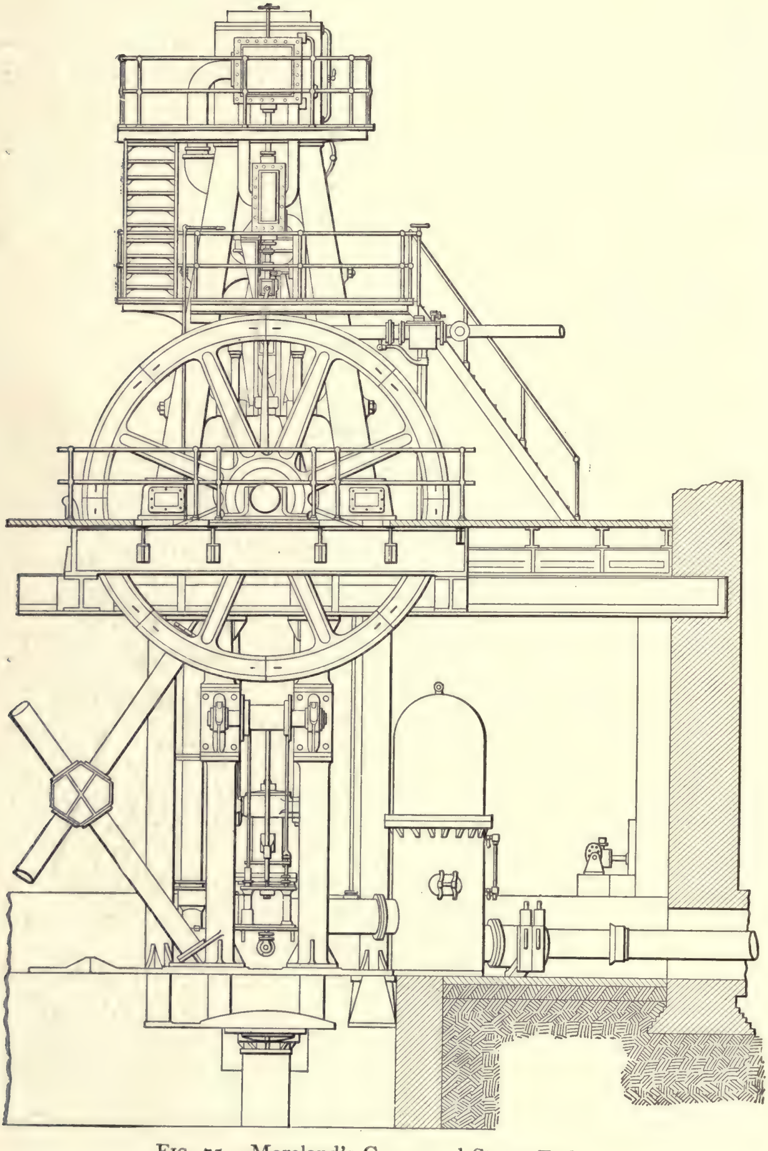


Fig. 75.—Moreland's Compound Steam End.

The exhaust from the low-pressure engine was carried to a jet condenser placed beside the engine, while the air pump was

placed below the main pedestal bearing. The pump was 10.52 inches in diameter with $2\frac{3}{4}$-inch rods. It was to lift 3,000,000 gallons per twenty-four hours against 270 feet head. It was of the inside-packed plunger type with 280 small valves.

In August, 1878, this engine was tested for ten hours per day for twelve days, with all coal and wood charged, and gave a duty per 100 pounds of coal of 104,357,654 when delivering 3,060,000 gallons. On October 3, 1878, a 24-hour test gave 133,522,060 foot-pounds. This was a remarkable result, but no more so than the annual duty of 123,656,000 foot-pounds for the year 1888.

In 1889 Professor Denton tested this engine and obtained a duty of 124,720,000 foot-pounds with hard coal and 127,-350,000 foot-pounds with soft coal. These results were obtained with boilers giving an equivalent evaporation of 8.88 pounds of water per pound of coal for the hard coal and 9.35 pounds for the soft coal. A higher duty could be credited to the engine if the usual amount of 10 pounds were evaporated.

The next important water-works engine was that used at the Pettaconsett Water Works of Providence, R. I. This was built by Corliss originally for Boston, but was not accepted for some reason. Its test for six days in May, 1882, gave a duty of 113,271,000 foot-pounds.

The inverted pump of Moreland and Thompson of 1868 proved to be so successful that in 1880 Moreland & Son designed a compound pump for the Eastbourne Water Works of England. This pump, Fig. 75, was of the close tandem compound steam type with a single-acting piston pump attached below. The plunger on the piston rod as shown in the earlier pump, Fig. 65, was arranged to give a discharge on each stroke. The general arrangement may be seen in the figure and reminds one of the lines of recent pumps. The valve gearing was of the Myer type and the high- and low-pressure cut off could be adjusted separately.

The general dimensions were as follows:

High-pressure diameter	20 ins.
Low-pressure diameter	38½ "
Pump-piston diameter	20 "
Plunger diameter	15 "
Stroke	40 "
Fly-wheel, 15 tons	15 ft. 7 ins. diam.

This pump was tested on February 18 and 19, 1884, by Messrs. Wallis and Borias, and showed a duty of 124,600,000 foot-pounds per 112 pounds of coal or 111,300,000 per 100 pounds of coal. It was running under a low head during this test and developed only 90.7 H.P., while it was designed to develop 160 H.P. The following data are given from the test:

Total revolutions in twelve hours	16,875
Average revolution per minute	23.44
Total amount of water, in gallons	759,456
Average total lift, in feet	243.94
" steam pressure above atmosphere	69.15
" vacuum	28.58 ins.
Horse-power water	77.97
Indicated horse power	90.70
Mechanical efficiency	86%
Coal consumed in twelve hours	1666 lbs.
Coal per horse-power hour of water	1.78
Coal per indicated horse-power hour	1.53
Duty per 112 lbs. of coal in ft. lbs	124,600,000
Duty per 100 lbs. of coal in ft. lbs	111,300,000

At this time the Holly Manufacturing Company brought out a high-duty engine to replace the quadruplex engine. This was designed by Mr. Harvey F. Gaskell, their superintendent. It was a simpler machine, and was the first standard design of high duty applicable to all forms of water works, the previous high-duty engines being of special design.

The first of these engines (Figs. 76, 77, and 78) was installed at Saratoga Springs, N. Y., in 1882.

From the figures it will be seen that the high-pressure cylinder is directly over the low-pressure cylinder and that the piston rods of these two cylinders are connected to the extreme ends of a massive walking beam by short connecting rods. This connection means that as one piston moves to the right the other moves to the left, each being at the dead

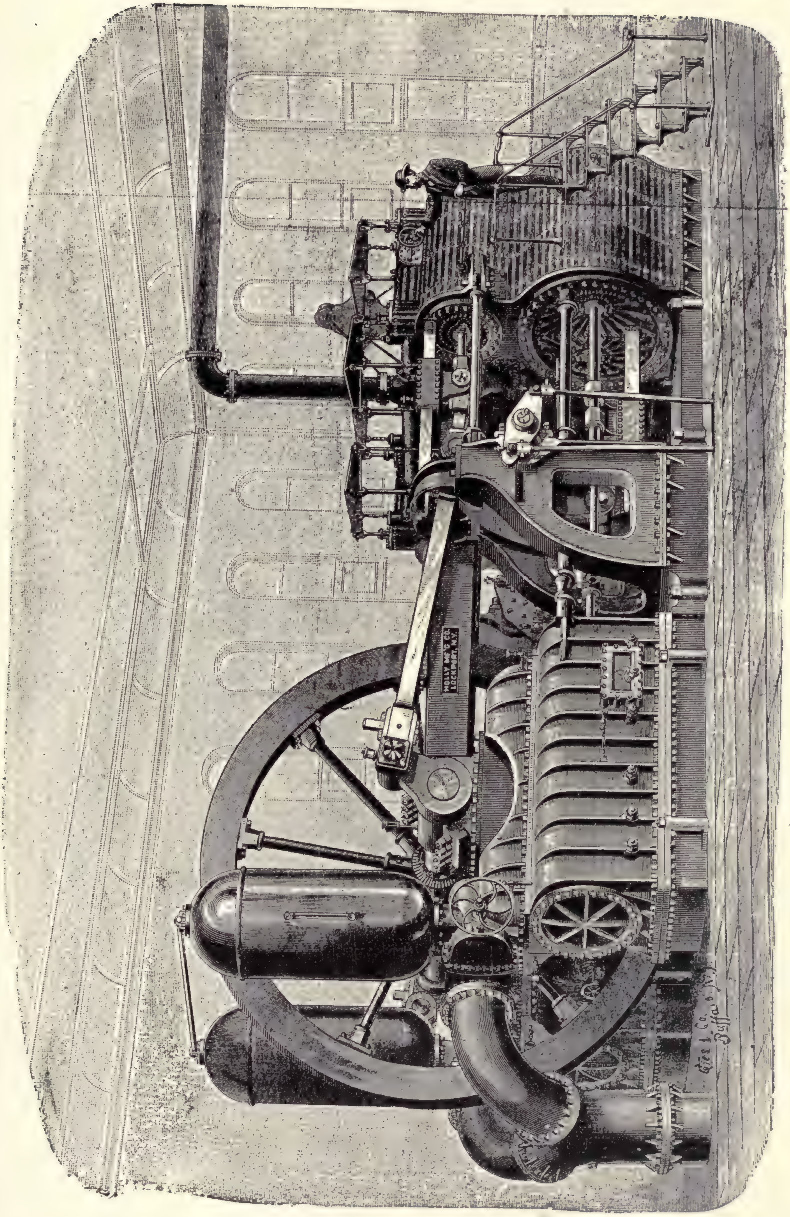


FIG. 76.—Gaskell Pump.

point at the same instant, and therefore it was of the Wolf compound type. This arrangement made it possible to dis-

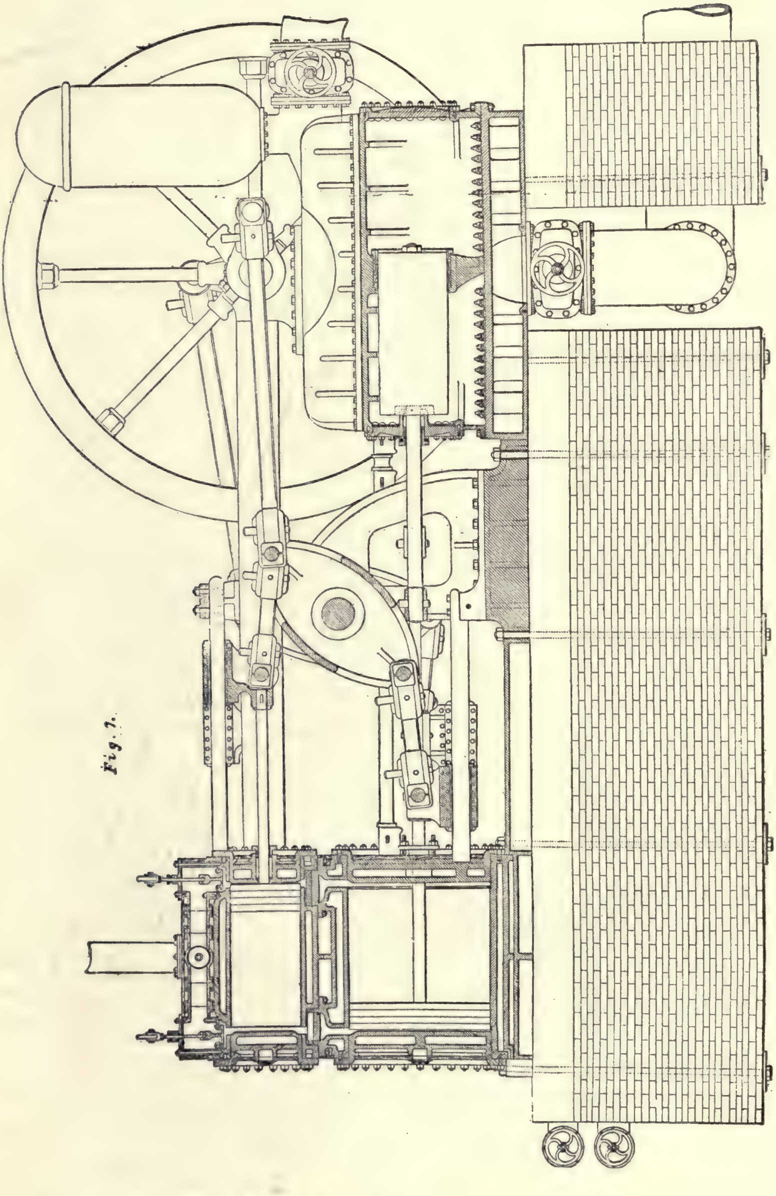
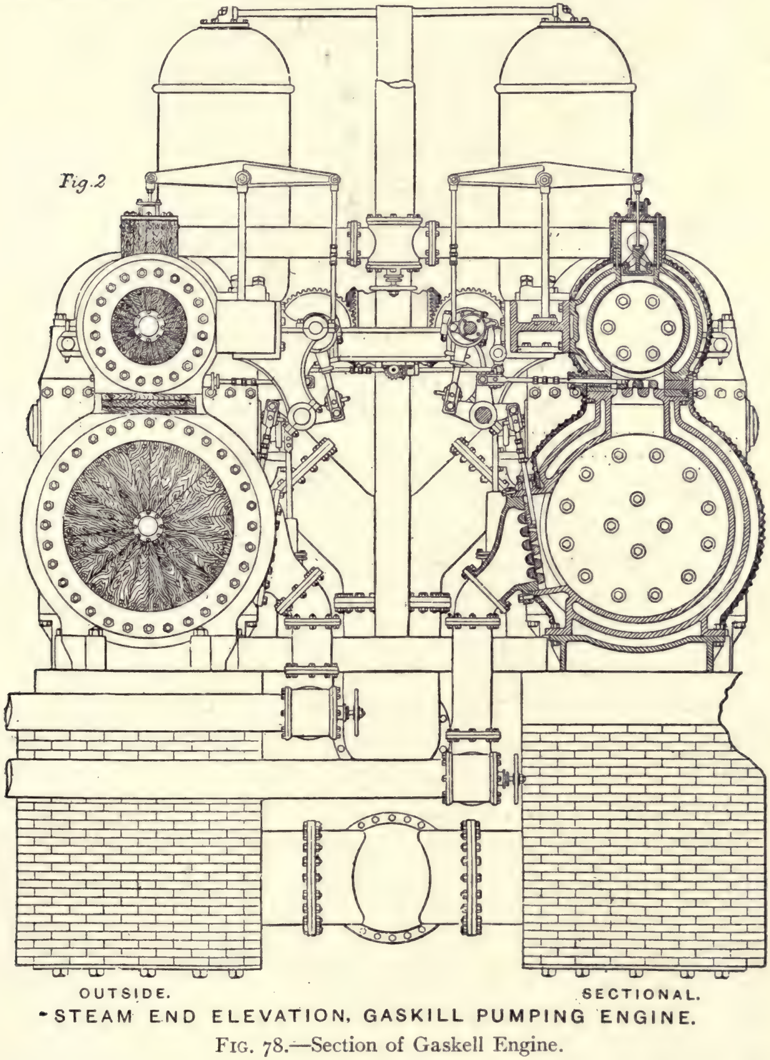


FIG. 77.—Gaskell Pump.

charge directly from the high-pressure cylinder to the low-pressure through the gridiron valve seen in Fig. 78.

A connecting rod from the upper end of the beam joined it to the crank of the fly-wheel shaft and by connecting the two sides or engines by cranks at right angles the motion of

Fig. 2



OUTSIDE. SECTIONAL.

STEAM END ELEVATION, GASKILL PUMPING ENGINE.

FIG. 78.—Section of Gaskell Engine.

the pumps was made steady. The valve gear is driven by two longitudinal revolving shafts driven by bevel wheels, the high-pressure steam valve being of the poppet type with an

automatic relief while the other valves are of the gridiron type. The condenser can be driven from a lever attached to the trunnions of the walking beam, although in the Saratoga engine a Buckley condenser was used. The water end consisted of an inside packed plunger, the Worthington system of a large number of small valves being used in it. In this engine there were 672 water valves. There were two tests made on this pumping engine; one in November, 1882, and the other in June, 1883; both of the results showed a duty of about 113,000,000 foot-pounds, although for a short period a duty of 127,000,000 was obtained. The dimensions of this engine were:

Diameter high-pressure cylinder	21 ins.
" low-pressure cylinder	42 "
" pump plunger	20 "
" " rods	4 "
Stroke	36 "
Capacity	5,000,000 gals.

There were subsequently many changes made in the details of this engine, but the general plan was unaltered. Charles T. Porter in his report of 1883 on the Saratoga engine makes the following statement: "In the details through which this general plan has been carried out, there are no features which seem open to criticism, but, on the contrary, all seem entitled to commendation. The construction is thoroughly mechanical in every respect. The forces are transmitted and the strains are resisted in the manner theoretically the most correct. . . . I do not think that the study of any machine has ever given me a stronger feeling of confidence in its durability." This shows the kind of designing done by Mr. Gaskell and the kind of work done by the Holly company.

While these pumps were being built in America, the European practice seemed to hold to the Cornish steam pump (Fig. 79), although in many cases horizontal fly-wheel engines were connected through other rods or gears to horizontal or vertical pump cylinders (Figs. 80 and 81), and in certain cases the beam engine was built with a fly wheel (Figs. 82 and 83).

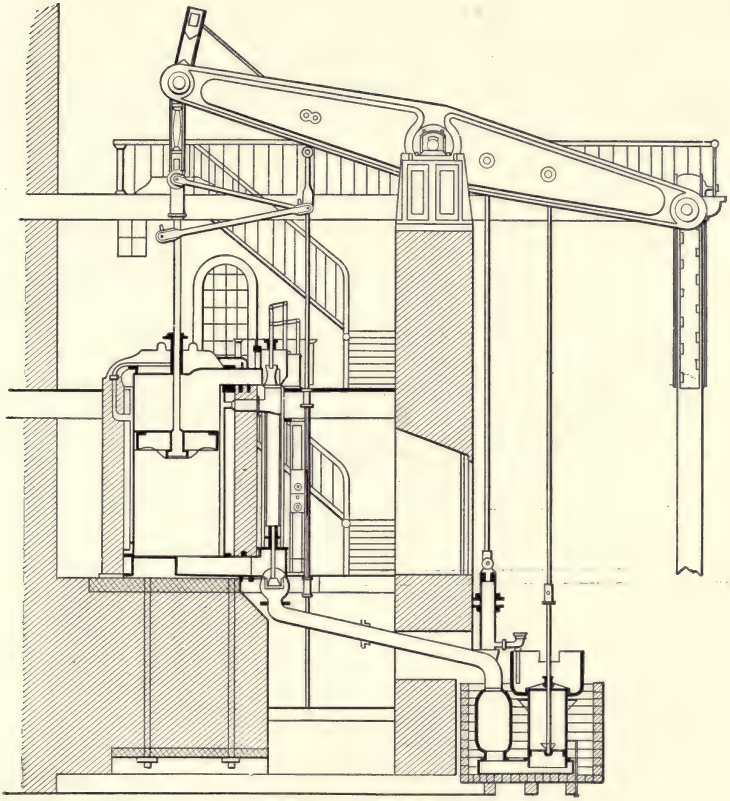


FIG. 79.—Cornish Engine of 1878.

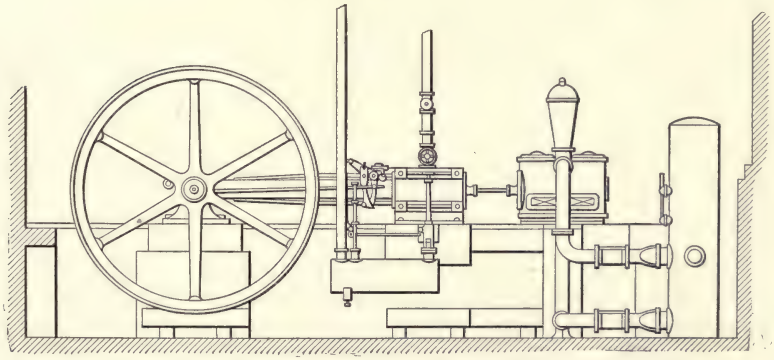


FIG. 80.—Horizontal Fly Wheel Pumping Engine.

The pump shown in Fig. 83 was one constructed by Simpson & Co., for the Lambeth Water Works of London. This pump was of the type used for many years by this company and gave unusually good results. The steam end consists of a compound engine connected to the beam by a Watt parallel motion. A fly wheel was used to replace the heavy bob weight.

The technical press of that time does not record any remark-

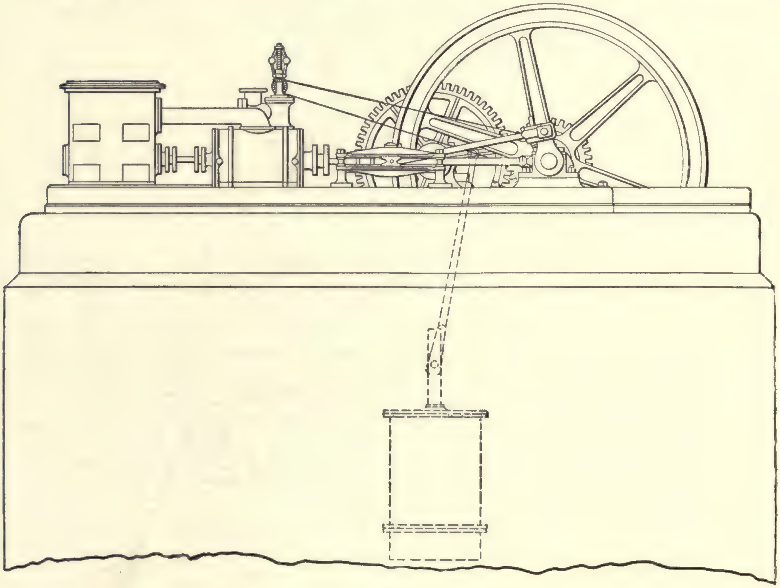


FIG. 81.—1874 Fly Wheel Pump.

able duties for these machines, and for that reason these are not described here, although the student who cares for examples is referred to the bibliography at the end of the book.

An exception to the statement above must be mentioned—the pump used for the city of Paris at the St. Maur station. Although installed in 1876, it was somewhat similar to one installed in 1871. They were built by Farcot & Sons. The steam cylinder was 39.4×70.8 inches, and the water cylinder was 14.2×70.8 inches. The distance from the center of the shaft to the center of the water cylinder was 50 feet 8 inches. The pump had a piston speed of 350 feet per

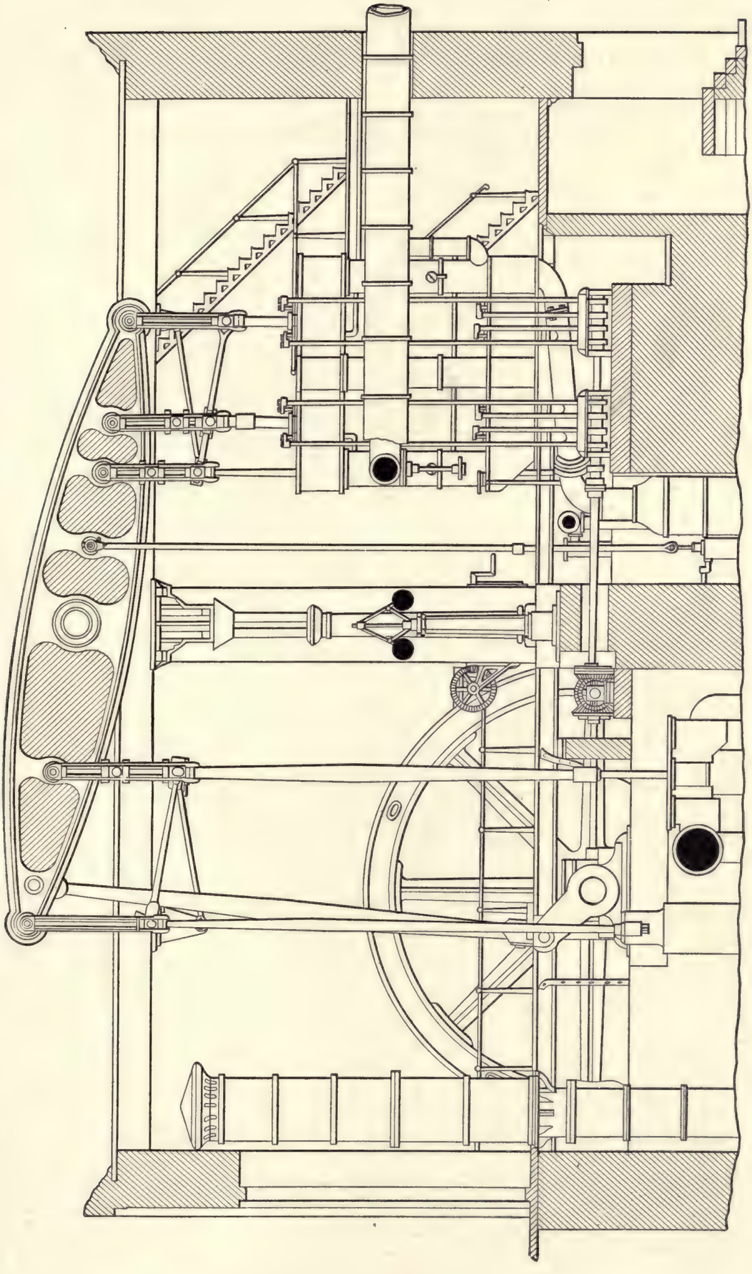


FIG. 82.—Berlin Water Works Engine.

minute, which meant about 30 R.P.M., a high speed for that day. At this speed it pumped 3,000,000 gallons in twenty-four hours. To get such a high speed the valves on the suc-

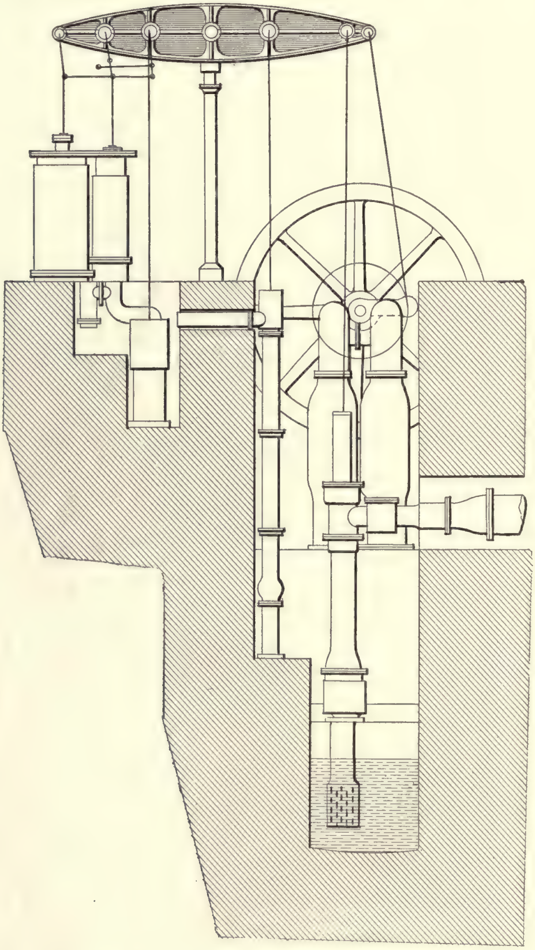


FIG. 83.—Simpson's Pumping Engine.

tion and discharge side were made so as to open simply and give a free passage to the water. This may be seen in the figure. The passages through the pump were made quite large so that the water would have a free path.

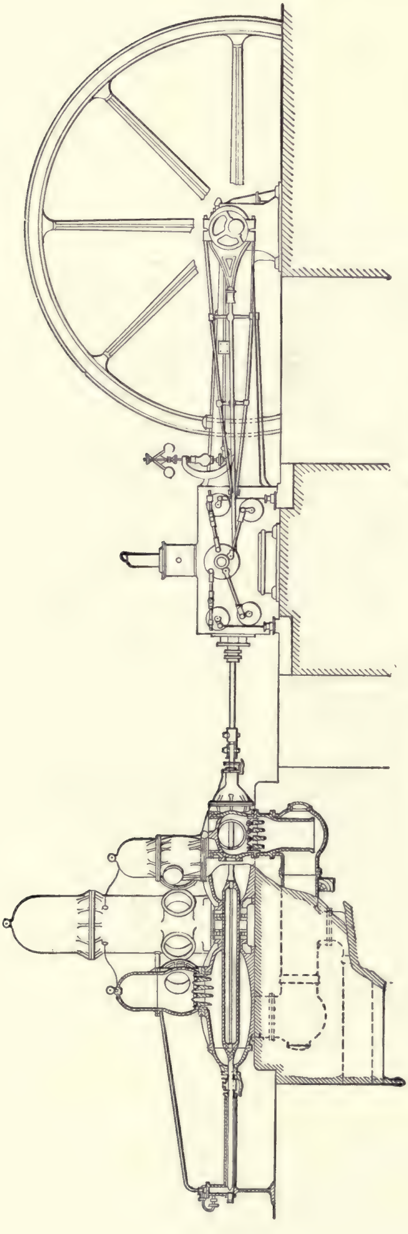


Fig. 84.—St. Maur Pump of Farcot & Sons.

The extended tail rod on the pump plunger was used to pump air into the air chambers to keep them charged.

The pump was driven by a single-cylinder condensing engine, with the air pump driven by a lever connected to the

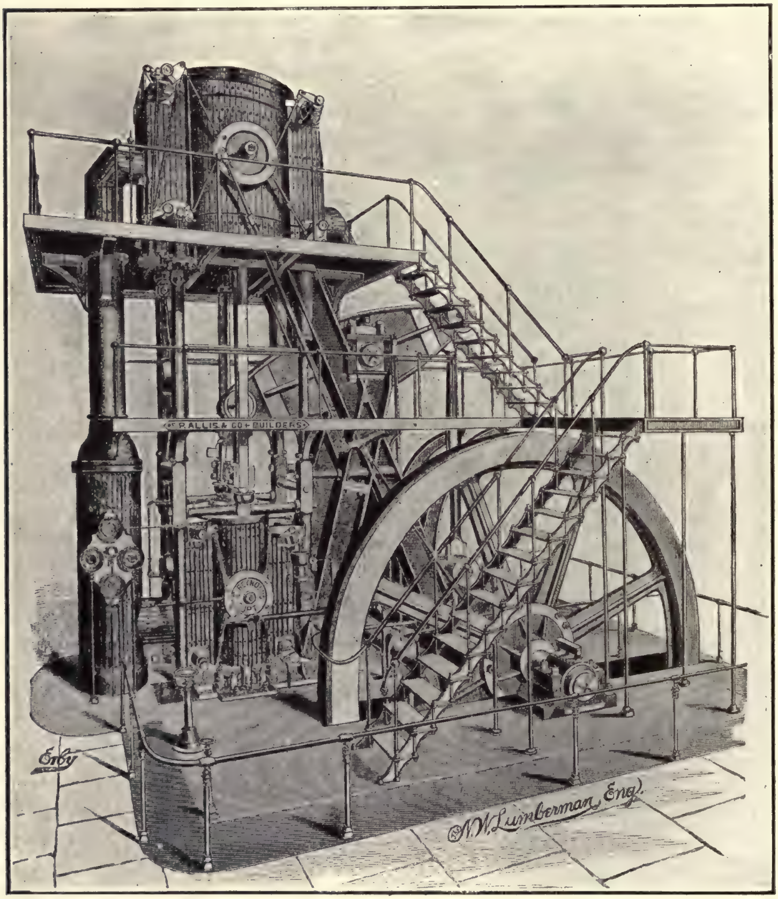


FIG. 85.

cross-head, and so well was the machine designed that the makers reported the development of an indicated horse power on 12.1 pounds of steam per hour or 1.54 pounds of ordinary coal while 2.03 pounds of coal were used per pump horse

power per hour. This would mean a duty of 78,000,000 foot-pounds per 100 pounds of coal.

The 6,000,000-gallon Reynolds pump of 1881 (Figs. 85 and 86) for Milwaukee shows a type similar to many of the European pumps of this period in its general arrangement of cylinders, except that there was an additional low-pressure cylinder

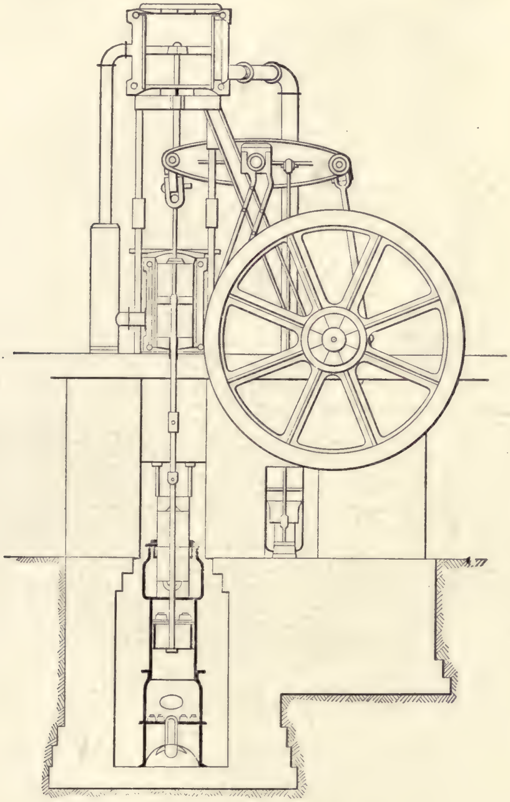


FIG. 86.—Reynolds Milwaukee Engine.

added above the beam. The duty of this engine was 104,000,000 foot-pounds per 100 pounds coal. Following this came the two 12,000,000 gallon Allegheny pumps of 1883 (Fig. 87), in which there were three cylinders to each pump—one high pressure and two low pressure. The engine was an entirely

new design for large engines, although small pumps had been arranged in this general manner much earlier. The cranks were placed at 120° and the number of working steam cylin-

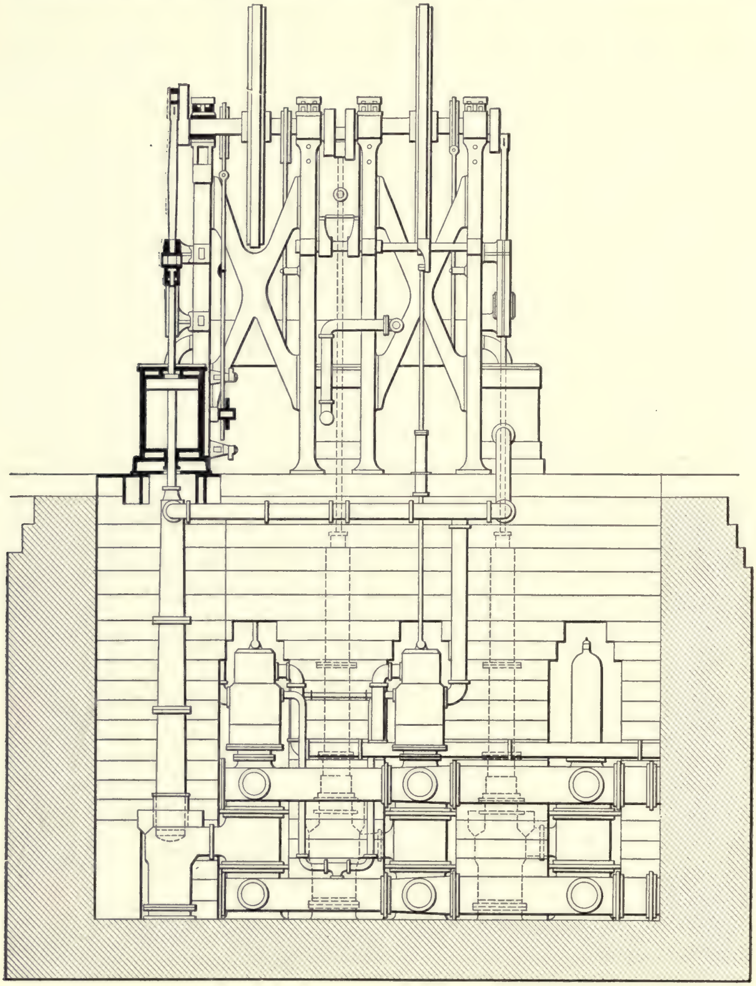


Fig. 87.—Allegheny Pump.

ders was reduced to three in place of four, so common in the standard type of pump of the day. The arrangement of the pump and steam cylinders is clear from the figure. The valve

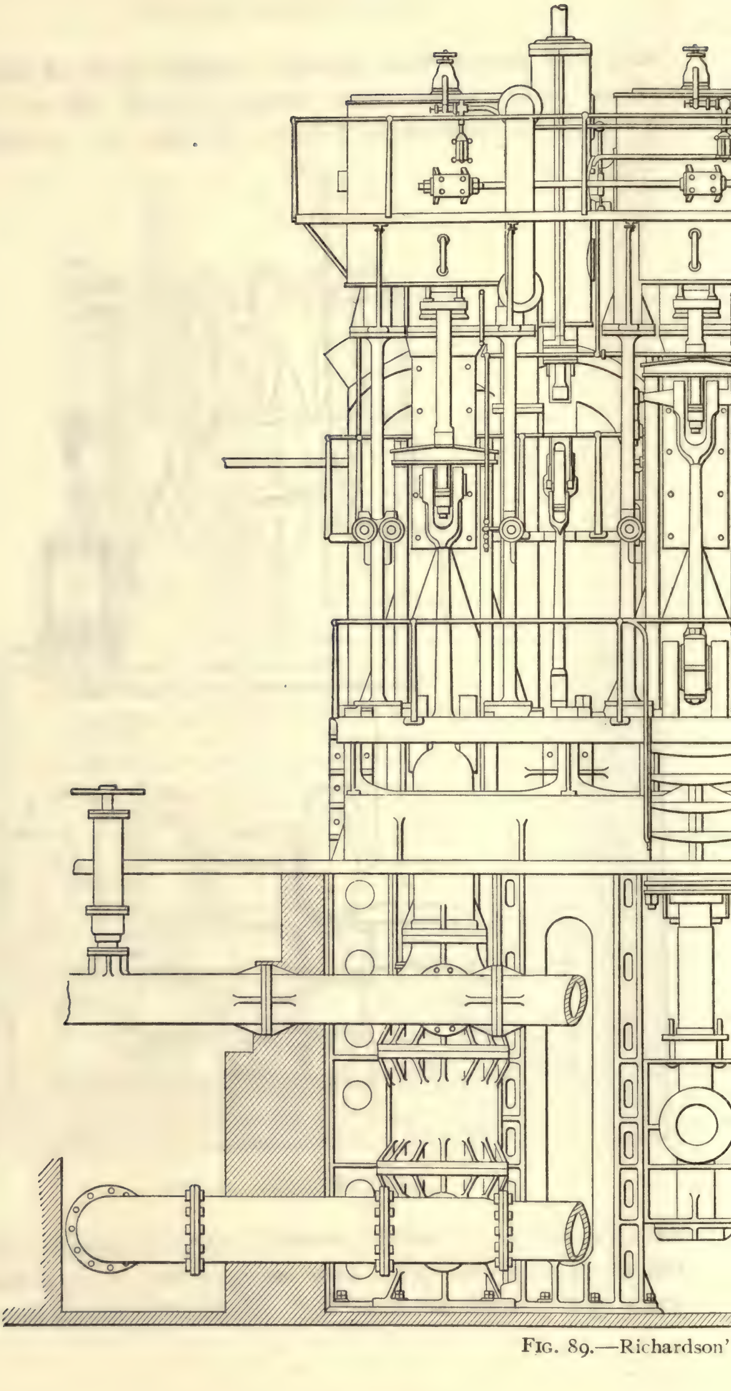
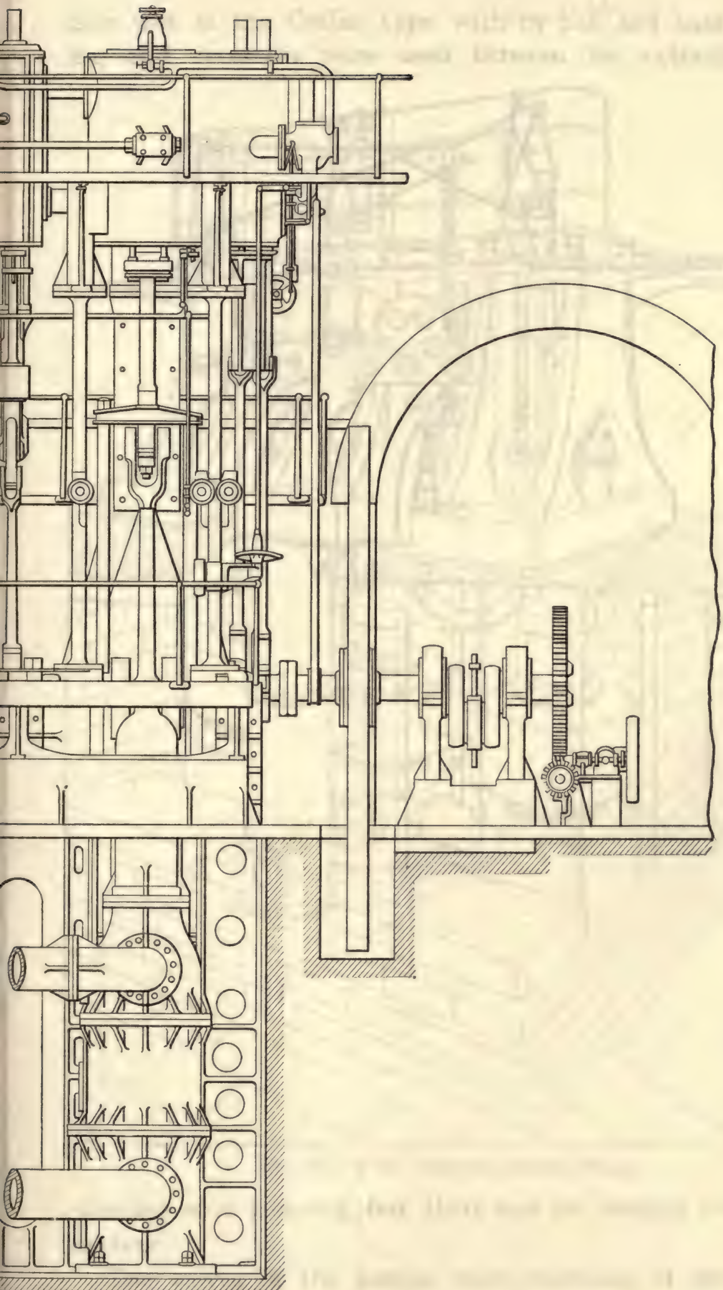


FIG. 89.—Richardson'



-expansion Engine.

(*To face page 94*)

gear was of the Corliss type with fly ball and hand governing and receivers were used between the cylinders. The

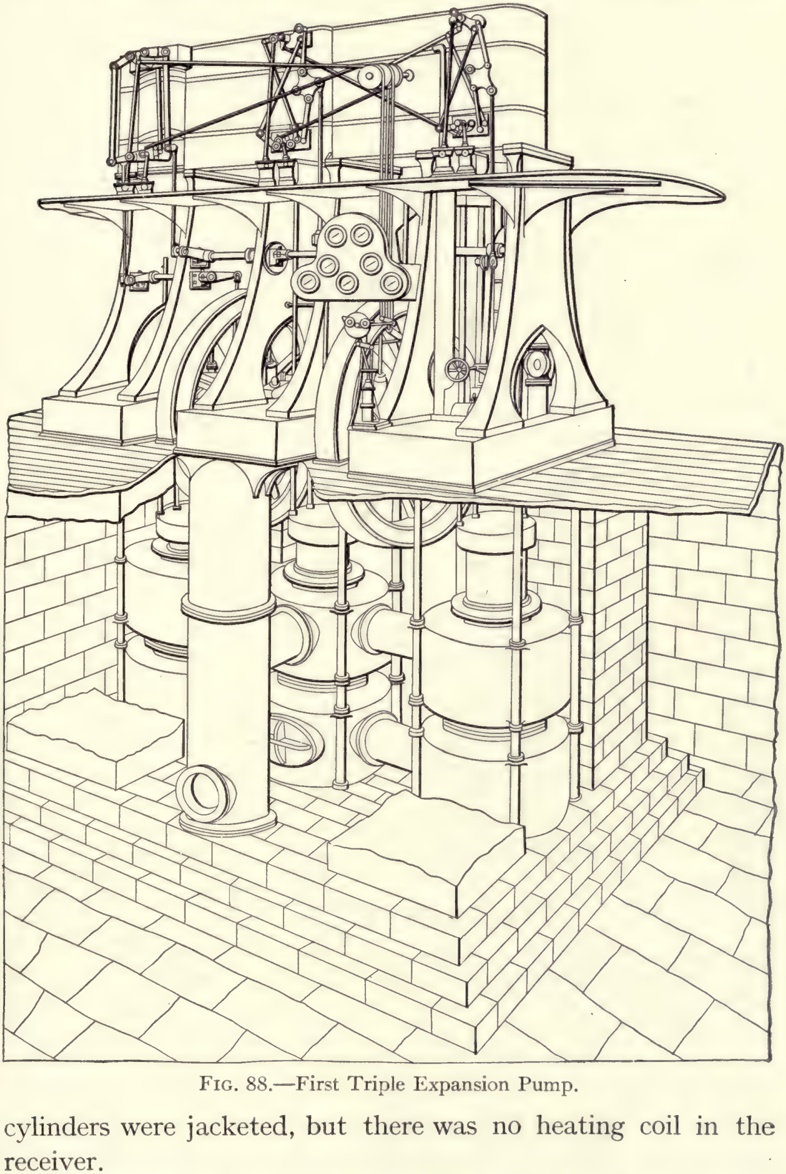


Fig. 88.—First Triple Expansion Pump.

cylinders were jacketed, but there was no heating coil in the receiver.

The valves of the pumps were originally of the Cornish

double-beat design, but these were later changed to small valves supported on a cage or basket placed over the opening for the Cornish valves.

The test of this engine gave 107,000,000 foot-pounds per 1000 pounds of steam when tested in 1884 by Mr. C. A. Hague and Professor David M. Green. It represented a type of engine of great value for small floor space, as was the case of the Eastbourne engine of 1880.

After the Allegheny engine of 1883 Allis built several compound-beam fly-wheel engines of about 2,500,000 gallons, and in one of them for Hannibal, Mo., the clearance was reduced by putting the Corliss valves in the head of the cylinder. This engine on test in November, 1885, by Mr. C. A. Hague, gave a duty of 118,327,025 foot-pounds per 100 pounds coal at a gauge pressure of 79 pounds. This engine then led to the design of the triple-expansion engine.

The triple-expansion engine (Fig. 88) is the first of the most popular type of large pumping engine, although many other forms have been suggested and used. This pump was built for the city of Milwaukee. The figure shows clearly the arrangement of the pump and engine. This resembles, to some extent, the design of frame used by Moreland in England. It is in reality the same arrangement as that used for marine engines with pumps added below the bed plate of the engine.

The engine gave the high duty of 122,483,204 foot-pounds per 100 pounds of coal with 80 pounds steam pressure. The dimensions of the engine were as follows:

High-pressure cylinder diameter	21 ins.
Intermediate-pressure cylinder diameter	36 "
Low-pressure cylinder diameter	51 "
Pump plunger	23½ "
Stroke	36 "
Capacity per twenty-four hours	6,000,000 gals.

The first triple-expansion pump in England (Fig. 89) was introduced in 1891 by S. Richardson & Sons, and gave a steam consumption of 13.53 pounds per I.H.P. hour. The use of

triple-expansion for direct-acting pumps was also introduced at this time by Davison in his water-works pumps. These gave good results and increased the duty of direct-acting pumps.

In 1885 Worthington brought out his high-duty engine,

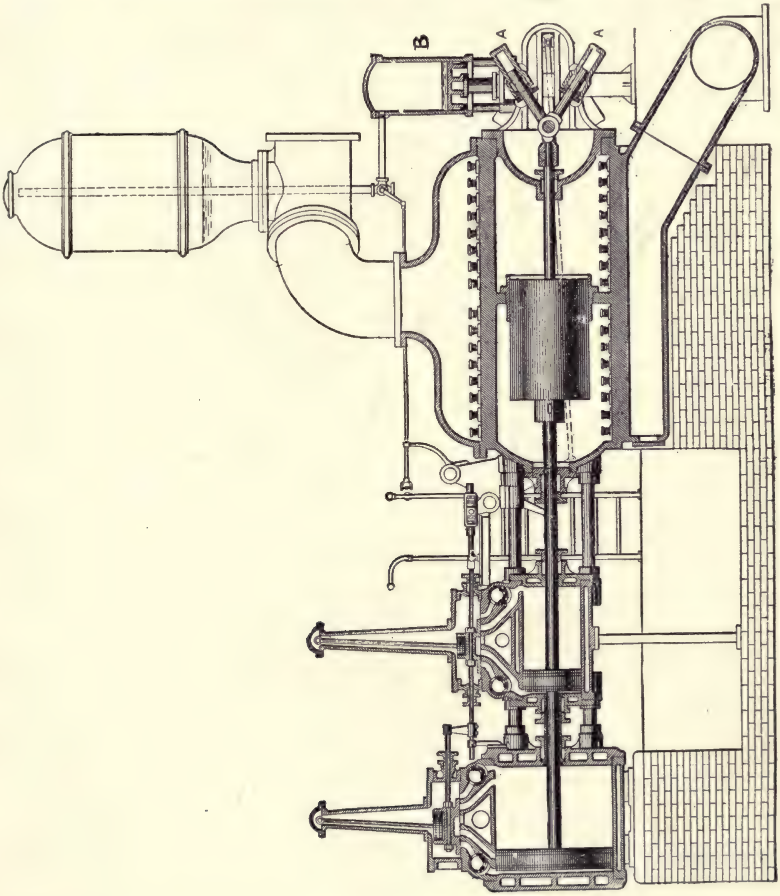


FIG. 90.—Worthington High Duty Pump.

built exactly as the duplex engines, but with the addition of compensating cylinders so as to use steam expansively. It was not possible to obtain the high duties of the expansive fly-wheel pumps with the ordinary duplex pump, and although these could be sold much cheaper, their duty was so low as to

prevent their use in many cases. The use of heavy reciprocating weights was not feasible, and Worthington improved the invention of Mr. J. D. Davies of 1879 by which steam could be used expansively. The arrangement as applied by Mr. C. C. Worthington is shown in Fig. 90, where, as the piston moves to the right, the tail rod of the pump forces the plungers into the oscillating cylinders *AA* against water pressure from the cylinder *B*, to which they are connected by pipes. The pipes are connected through stuffing boxes to the trunnions

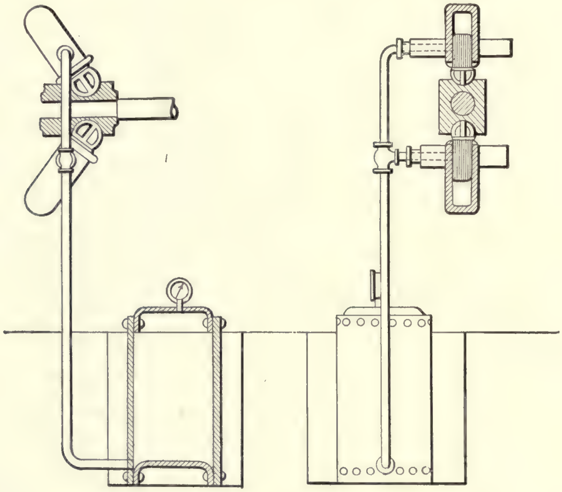


FIG. 91.—Independent Air Tank Compensator.

of the cylinders. It will be seen that the side thrust from these is balanced on account of their symmetrical position. The cylinder *B* is connected to the top of the air chamber in the force main of the pump, and thus the pressure of the air cushion above the fluid in *B* is always equal to that in the force main. By the arrangement of multiplying cylinders at the base of *B* the unit pressure on the plungers of *AA* is much greater than that of the force main, as the area of the plunger is much smaller than the piston of *B*.

The pistons are opposed by the pressure in the compensators

until the engine reaches its mid-position; the plungers are here in a vertical position. From this point to the end of the stroke the pressure on the plungers is aiding the motion, the exact amount being proportional to the cosine of the angle of inclination to the horizontal. This then means that at the beginning of the stroke the steam pressure will have to exceed the pressure on the main pump plunger, diminishing until the mid-stroke, and from there on it may be less than the pressure on the main plunger. This permits the use of steam expansively. The application of an independent air tank (Fig. 91) is used at

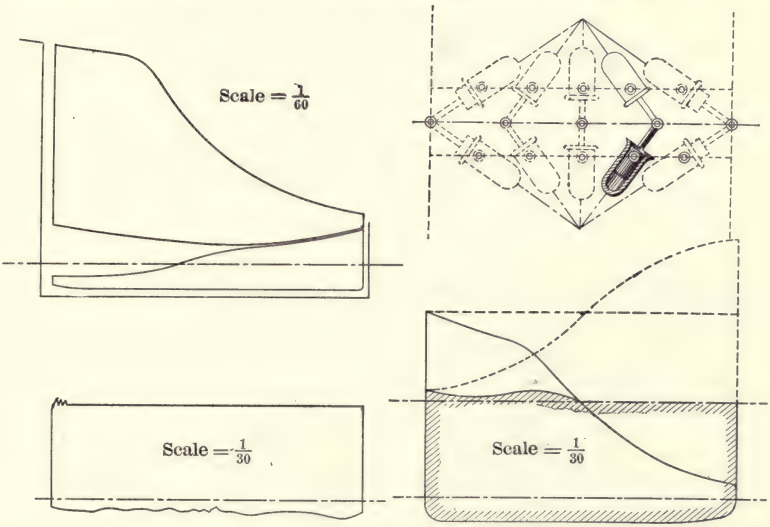


FIG. 92.—Action of Compensator.

times with this system. The figure gives a clear idea of the arrangement of the rams and plungers.

To better illustrate the action of this, Fig. 92 is given; in it the indicator cards, the pump cards, the various positions of compensating cylinders and the combined cards are shown. At the lower right-hand side of the figure the total steam-pressure card has been placed over the total water-pressure card and a dotted line has been drawn showing the variation of total pressure exerted horizontally by the compensators. It is evident that during the first half of the stroke the excess

of total steam pressure just equals the positive resistance from

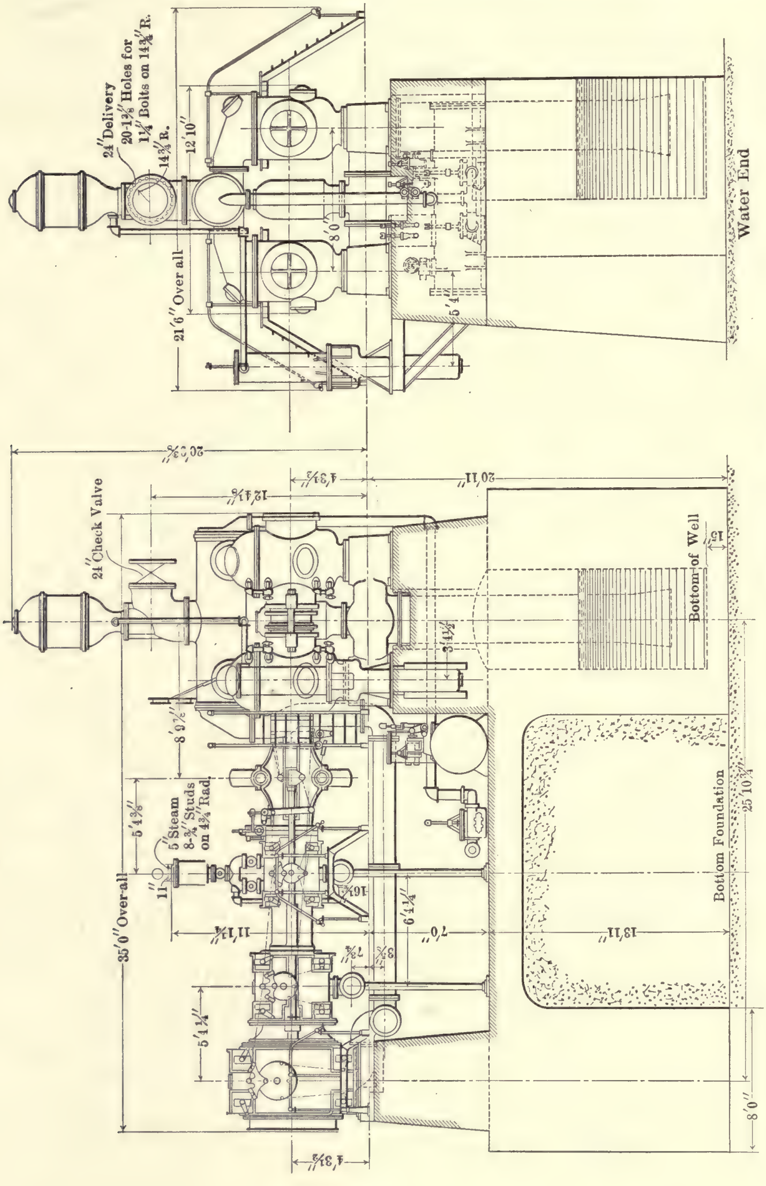


FIG. 93.—Fall River Worthington Pump.

the compensators, while during the latter part, the excess of

water pressure is made up of the negative resistance or returned energy from the compensators. This enables the pump to obtain all of the advantages of the expansive use of steam and tests of these engines have shown this.

Fig. 90 shows the application of this in 1885, wherein the forms of water end and steam end are obvious. The steam end is jacketed throughout and has separate steam and exhaust passages with rotary cut-off valves in addition to the balanced slide valves. This type was soon changed so far as the valve was concerned, but the compensating principle was practically the same.

A test of the first engine of this type built in 1885 for the water works of New Bedford, Mass., gave a test duty of 79,238,160 foot-pounds on 100 pounds of coal, and this test was followed by another by Mr. John G. Mair, a representative of Simpson & Co., the English builders of pumping machinery. Mr. Mair found that the engine used from 14.53 pounds to 15.05 pounds of steam per I.H.P. hour.

This engine was of the compound type, as shown in the figure, and was developed extensively, being used in a vertical position in places where ground area was limited, as in an installation made at Memphis, Tenn. This same engine was tested in 1891 and gave a duty of 117,325,000 foot-pounds per 1000 pounds of steam.

This compensated form of pump was also developed into a triple-expansion steam end. One of the first built by Worthington in 1891, for Turtle Creek, Pa., was of the vertical form with the high-duty attachment. The triple-expansion forms are shown in Figs. 93 and 94.

Fig. 93 shows the pump installed at Fall River, Mass., for the city pumping station in 1909. It has three steam cylinders, 21, 33, and 60 inches in diameter, respectively, arranged in tandem with the 24½-inch water cylinder. The common stroke is 36 inches. The high-duty attachment is placed between the high-pressure steam cylinder and the water cylinder.

The pump takes its suction from an open conduit and pumps directly into the city mains at about 100 pounds per

square inch. The capacity was 10,000,000 gallons per twenty-

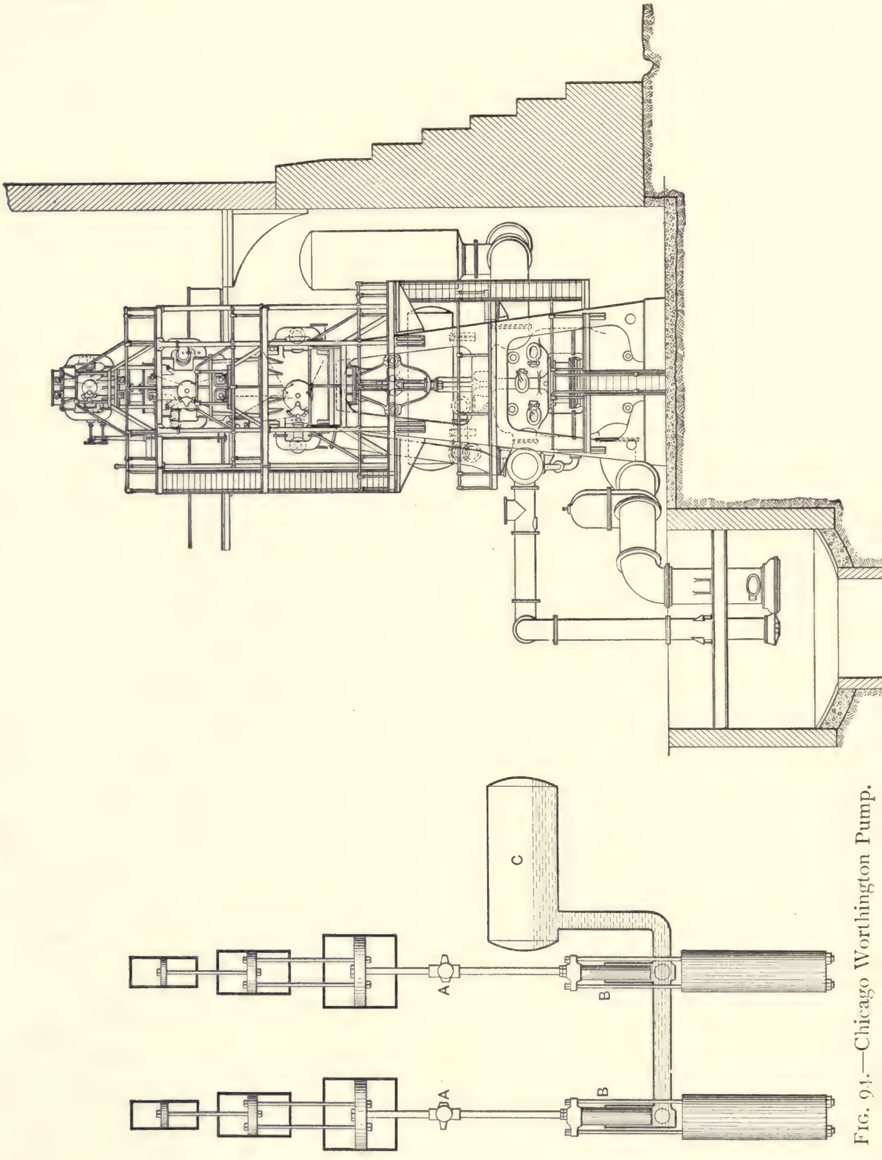


FIG. 94.—Chicago Worthington Pump.

four hours. The pump is of the outside-packed plunger type

with special glands. The cylinders are jacketed and reheater coils are used between the cylinders. The auxiliary pumps for charging the air chamber of the compensator, the air pump, and jacket pump are driven from the main pump. The stroke governor regulates the stroke, keeping it at its full value by means of a change of the pressure in the compensator.

The official test of this pump gave a duty of 136,500,000 foot-pounds per 1000 pounds of dry steam. A pump similar in principle to the above was installed at Montreal in 1908 and gave a duty of 177,538,000 foot-pounds per 1000 pounds of steam.

Fig. 94 gives a good idea of the vertical form as installed at the Central Park pumping station in Chicago, although the figure is not of this particular pump. These pumps are duplex with steam cylinders, 21, 33, and 60 inches in diameter, and of 50-inch stroke, while the double-acting water cylinders are $34\frac{1}{2}$-inch diameter and 50-inch stroke. The steam cylinders are jacketed on heads and barrels and the valve gearing is of the four-valve rotative type. At 18 revolutions per minute this pump lifted 20,000,000 gallons per twenty-four hours. The duty on test was 174,735,801 foot-pounds per 1000 pounds of superheated steam used.

By means of the balancing plungers shown in Fig. 94, the weight of the moving parts is supported by plungers *BB*, which force water into a tank *C* against air pressure. The pressure in this tank is regulated by the pressure in the discharge main. When this pressure falls, due to a break in the line, the weight of the moving parts would not be supported by the water pressure and the pump would soon come to rest. The steam pressure is used to force the water on each stroke. At points *AA* on the piston rods between the low-pressure cylinder and the water cylinder the digh-huty attachment is applied which permits of the expansive use of steam, as shown previously.

The suction pipe, which is connected to a large cistern in the pump room, is joined to the surface condenser so that all of the water passes through the tubes. In this way the condensation is cared for in a station situated in the center of the city where condensing water is not at hand.

One of the latest types of compensating, direct-acting pump, permitting the expansive use of steam without the use of a fly wheel, is the invention of Mr. Luigi d'Auria. The pump (Fig. 95) has a cylinder *A* between the steam and water cylinders, the ends of which form the extremities of a pipe circuit which in some cases is used as the base of the pump. The cylinder *A* contains a piston, and as this moves back and forth the water with which the system is filled is compelled to travel first in one direction and then in the other.

At the beginning of the stroke the inertia of the water

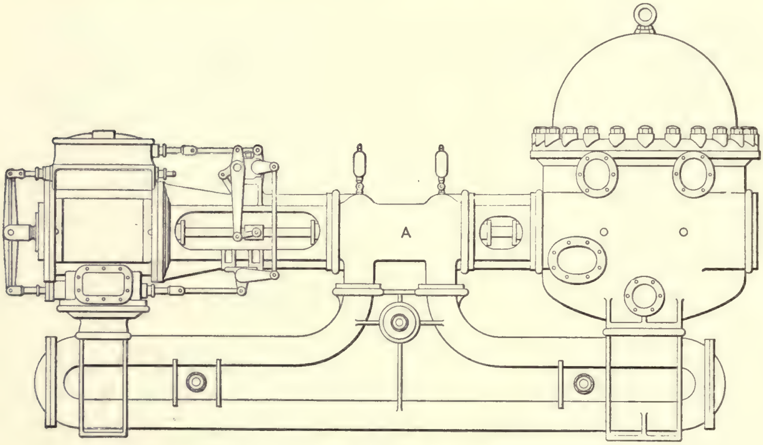


FIG. 95.—d'Auria Pump.

utilizes the excess steam pressure, while in bringing the water to rest it will exert a force to supply the deficiency. In this way steam may be used expansively in the steam cylinder to drive the pump against a uniform water pressure. The steam may be used in one or more cylinders, depending on the purpose of the engine. When multiple expansion is wanted it can be applied to this engine. The compensation is so perfect with this machine that high speed may be employed with no danger of pounding.

This method has also been applied to triple-expansion pumps where high duties have been sought.

About the end of the last century Mr. Charles L. Heisler of Erie, Pa., introduced a new type of compensated engine which he described in 1900. His design employs a duplex arrangement of cylinders, as shown in Fig. 96, which is built for triple expansion. The line drawings show that each piston rod is connected to a vibrating radius rod *A* by a short connecting rod *B* and that the radius rods are joined by the link *C*. Just before the left-hand engine reaches the end of its upper stroke, as in position 1, the movement of the link *C* in compression lifts the left radius rod and aids in the movement of that pump. Position 2 shows that the left pump is being aided by compression in the rod *C*, while the other positions show the pumps

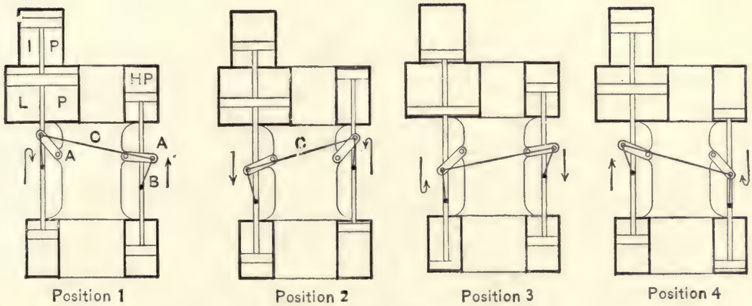


FIG. 96.—Heisler Pump.

at the two remaining starting points. In each of these positions the excess of pressure at the beginning of the stroke is carried over to the other side and aids the steam when that pressure is less than the resistance of the water. This method gave fair results and the builders claim duties of 130,000,000 to 160,000,000 foot-pounds per 1000 pounds of steam, although the author has no records of any tests giving such high duties.

The method of having one side of the pump aid the other by means of a linkage was not new at this time, for in 1874 Fielding of England proposed such a scheme and in 1887 Henry Davey patented a less complex form. The Davey method of compensation is given in Fig. 97. In this system single-acting water plungers are connected with double-acting steam cylinders.

When the plunger in *A* is moved to the right there is little resistance to its motion, so that the excess of steam pressure is carried through the rod *D* to the plate *C* and from it to the other side of the pump by the rod *E*. At the beginning of the stroke, when little of the pressure from the side *A* is needed, it is seen that the rod *D* has little leverage about the center of *C* while *E* has a long leverage. When, however, the plunger of *B* has been driven to the left and the steam on that side has expanded so that more pressure is needed, the leverage of the rod *D* becomes greater, giving more pull on the rod *E*. At the end of the stroke the low pressure on the side *A* is pulling with full leverage and exerting a great force on *E*, which pulls on

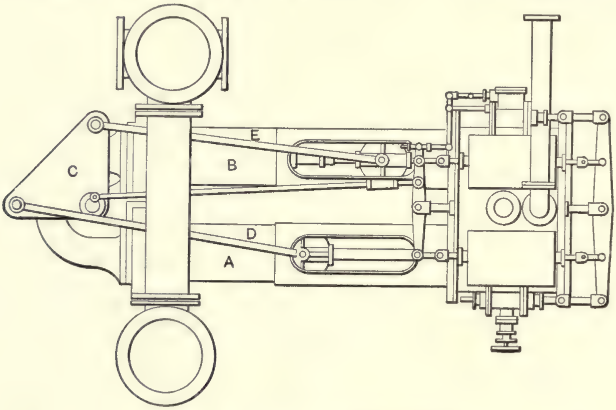


FIG. 97.—Davey Compensator.

the plunger of *B*. *B* requires this additional pressure, as the steam on that side has expanded to a low pressure. The valve gear is such that these events can take place as described, the two pumps reversing at the same time. The pumps described by Davey in London *Engineering* in 1877 and the Watt engine described in the same magazine in 1885, are mentioned as similar to the above in principle.

From the introduction of the triple-expansion fly-wheel pump of Allis in 1886 to the present there have been a number of improvements in the arrangement of steam valves and reheaters, pump cylinders and valves, condensers and other details. With the gradual increase in the steam pressure, the

duty, when measured on the basis of 1000 pounds of steam, has increased together with that of the duty on the basis of 1,000,000 B.T.U. The type first used by Allis has been adopted by other large engine builders and this design in the hands of the Worthington, Blake, Holly, Snow, and Southwark companies has proven it to be a good one. Mr. E. D. Leavitt, Jr., has also designed special types during these years which have given excellent results, as will be seen later. The duties on the basis of 1000 pounds of steam have gone from 122,452,729 foot-pounds in 1886 to 154,048,704 foot-pounds in 1892, to 179,454,250 in 1900, and finally to 181,048,605 foot-pounds in 1906. This method of stating the duty has the objection that the efficiency is dependent on the amount of heat in each pound of steam supplied. The amount of heat in 1000 pounds of dry steam depends on the pressure of the steam and consequently it is perfectly possible to have an engine with a smaller duty per 1000 pounds of steam more efficient actually than one with a a higher 1000-pound duty. The method of expressing the duty as the number of foot-pounds per million British thermal units supplied is far better, as in this case the real efficiency is being measured.

The engine giving the highest actual thermal efficiency, according to C. A. Hague, is the Norberg quadruple-expansion engine of 1899, which gives 22.80 per cent; however, the thermal efficiency based on useful work delivered by the pumps, obtained by Professor Thurston in 1899, was 20.9 per cent. This result was within 84 per cent of the theoretically perfect engine. This pump was built for the Pennsylvania Water Company, near Pittsburg. There were four steam cylinders as well as the pump barrels and fly wheel. The pump was built on peculiar lines because the water level of Allegheny River is subject to much variation, and moreover the pump was to be placed with two other similar 6,000,000-gallon pumps within a well of 38 feet diameter belonging to the company. The dimensions of this engine are as follows:

Diameter	high-pressure cylinder	19½ ins.
"	first intermediate cylinder	29½ "
"	second intermediate cylinder	49½ "
"	low-pressure cylinder	57½ "
"	plunger (double-acting) cylinder	14¾ "
Stroke		42 "
Fly-wheel diameter		13 ft.
Capacity		6,000,000 gals.

This engine gave the highest duty on the absolute basis, although it does not give so high a value per 1000 pounds of dry steam as those which have been built later; the best accepted duty on this basis being about 180,000,000 foot-pounds.

Figs. 98 and 99 illustrate two modern forms. The first is a pump installed in 1909 at Brockton, Mass. The steam cylinders are 19, 36, and 54 inches in diameter, and the stroke is 36 inches. At 40 R.P.M. the 36-inch single-acting plungers will lift 6,000,000 gallons per twenty-four hours against 300 feet head. The duty of this pump was 169,982,000 foot-pounds per 1000 pounds of dry steam at 150 pounds pressure.

The second pump illustrates a horizontal cross-compound condensing crank and fly-wheel Snow engine for Elyria, Ohio. The capacity of the double-acting plungers, 14½ inches in diameter, is 5,000,000 gallons per twenty-four hours at 36 R.P.M. The steam cylinders are 22 and 48 inches in diameter with a 36-inch stroke. The steam pressure was 120 pounds per square inch and the total water pressure happened to be the same. The duty was 137,000,000 foot-pounds per 1000 pounds of dry steam.

These two pumps illustrate clearly the form of modern high-duty pump for water works. Fig. 98 illustrates the vertical self-supporting pump in which the valve boxes and air chambers form the base for the pump, and the pump barrel is a separate casting. The steam valve gearing is of the Corliss type. The valves are placed in the sides of the high-pressure cylinder, in the sides and heads of the intermediate, and in the heads of the low. The side straddling rods leading from the cross-head to the end of the plunger so as to clear the shaft and crank are shown. The governors, valve controls, and oiling devices are also apparent. Such pumps are usually placed in basements so that the level of the main floor of the station is just over the air chamber.

Fig. 99 is the Snow type of norizontal pump designed to take up little floor space and yet have the cylinders for the

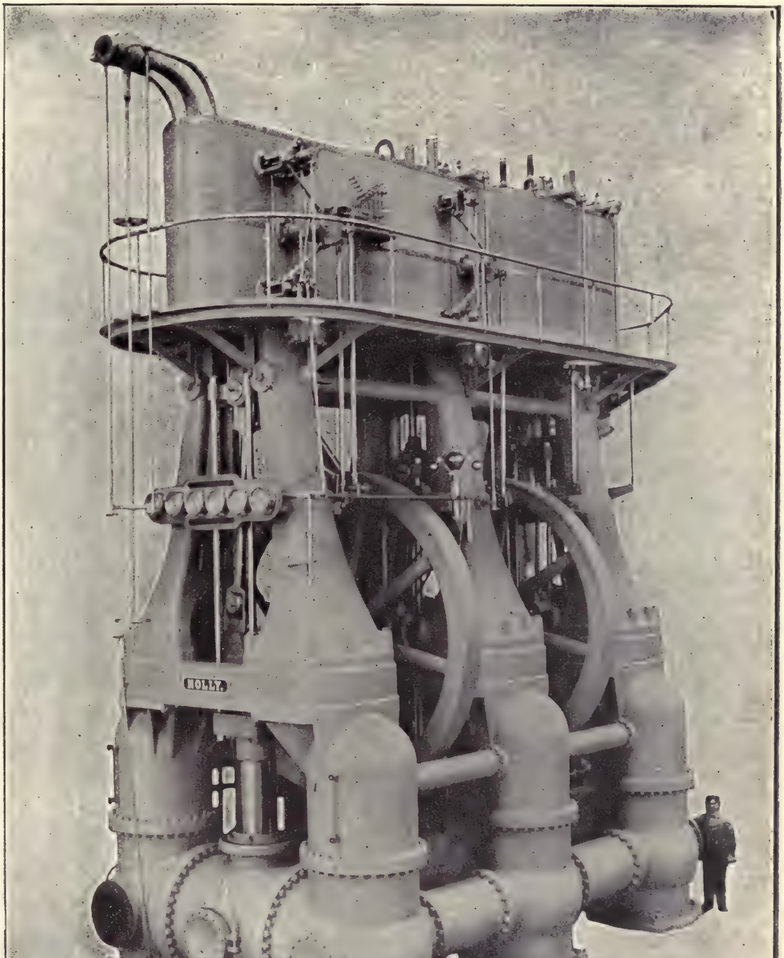


Fig. 98.—Holly Triple Expansion Pump.

water and steam ends in such a position that they may be easily examined and repaired. The method of clearing the

crank and shaft by two rods joining the cross-head to the plunger rod is clearly shown as well as the arrangement of air chambers, pump cylinders, steam receiver, and valves.

In this period of the important development of water-works pumping engines strides were being made with the centrifugal pump. As will be remembered, it was about the year 1846 that Andrews showed that curved vanes were better than the straight vanes in the old Massachusetts pump (Fig. 39). This development was carried out by Gwynne of England. At this time Andrews also suggested the value of having the vanes of

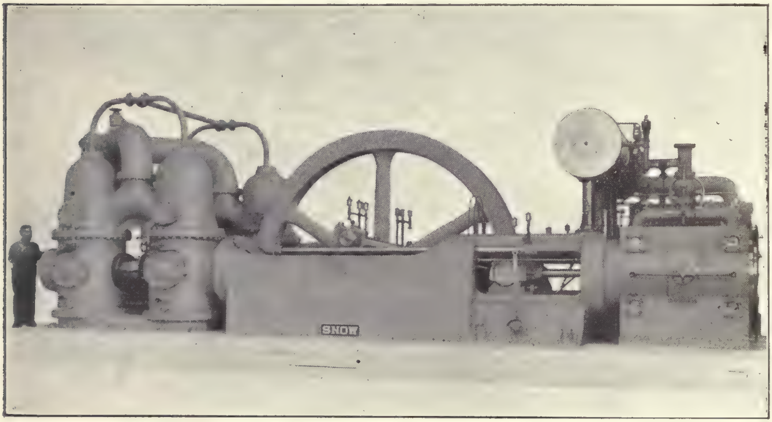


FIG. 99.—Snow Compound Pump.

the runner inclosed between two discs as shown in Fig. 40. In 1851 Appold showed the advantage of the curved vane of Lloyd's fans experimentally.

From now on the centrifugal pump was used extensively and the patent gazettes are filled with the record of improvements. The great field for this pump was the lifting of large quantities of water through small distances, where there was not much floor space available, and also where the original cost of the apparatus was of some importance. The firm of John & Henry Gwynne in England was the best known of the early builders, and their pumps for the drainage of the low-

lands of Holland and Denmark and the emptying of large drydocks were very successful.

Not only did the Gwynnes try low lifts of 15 and 30 feet, but in 1868 they built a pump with a runner 2 feet in diameter to lift water against a total of 18 feet suction and 114 feet discharge. The pump was driven at a speed of 910 revolutions per minute.

In 1869 these pumps were applied to condensers of steam engines. For this service they are admirably adapted, as a large body of water has to be lifted a small distance in the case of stationary engines, while for marine practice the only work the pump does is to overcome the resistance due to the moving of the water and the friction of the pipes and tubes.

The early study of these pumps was quite meager and their design was more or less empirical, but as their use extended a better understanding of them was had. Although inexpensive for the quantity of water handled and apparently of a form to give good results the efficiency of this machine was low. The earlier efficiencies of 50 per cent were increased to 65 and 70 per cent in 1885. Some records give as high as 75 per cent for this type of pump. High efficiencies can be had only by careful design and construction.

The application of this pump to other services increased. In 1869 it was employed for dredging, although a patent of Louis Schwartzkopff, granted in England in 1856, foreshadowed this application of this simple machine, while for draining wells vertical shafts were used in 1884. Before this time, however, the manner of balancing end thrust by bringing in water from each side was employed. The possibility of handling large quantities of water is illustrated by a pump built by R. Moreland & Sons for a dock at Malta, where it was required to lift 30,000 gallons of water per minute against a comparatively low head. This was built in 1887. Another pump was built a little earlier by the Southwark Foundry and Machine Co., designed to raise 40,000 gallons per minute.

W. O. Weber showed by experiment that the efficiency of a pump decreased after a certain head was reached, and so when

greater heads were to be overcome by centrifugal pumps, it was suggested by some (W. A. Booth claims the honor of this) that two pumps be placed in a series (Fig. 100). This then led to the multi-stage pump, where several pumps were brought together in one casting. The water discharged from one stage passes over to another stage, where its pressure is increased. Continuing in this manner the pressure against

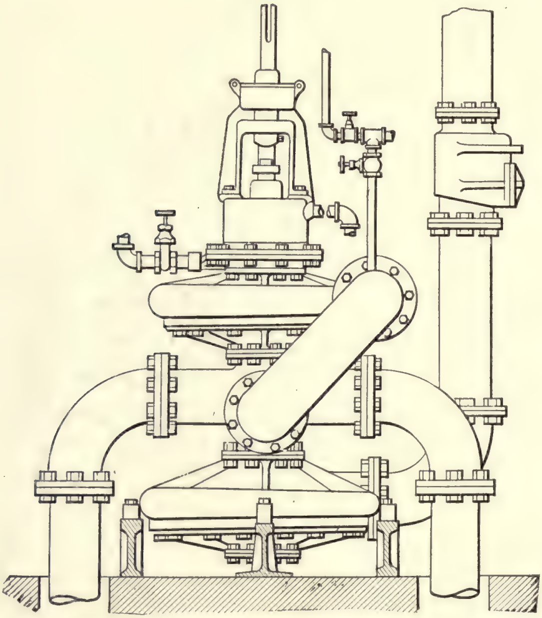


FIG. 100.—Two-Stage Pump.

which the pump will act effectively can be made as great as desired. The Allis-Chalmers Company have recently installed a five-stage pump which lifts 3000 gallons of water per minute against a total head of about 700 feet. Further details of the centrifugal pump will be considered in a later chapter.

The rotary pump has been the subject of many patents during this later period. The principles involved, however,

were all embodied in the earlier designs. A few will be described to give some idea of the progress made in this form of pump.

The Behrens pump of 1867 (Fig. 101) represents a form of rotary pump using rotating lobes with no sliding parts. The shafts *AA* are geared to move in opposite directions with the same velocity. The pistons *B* are so formed that they slide over the bore of the main cylinder *C* and the inner abutment liners *D*, which are stationary. These liners are carried on the heads on one side of the cylinder, while the pistons *B* are carried from the shaft *A* by means of a flange cast with the pistons. This is seen better in the small perspective view.

On turning the pistons in the directions shown in the figure, water is drawn into the space *E* while water in the space *F* is

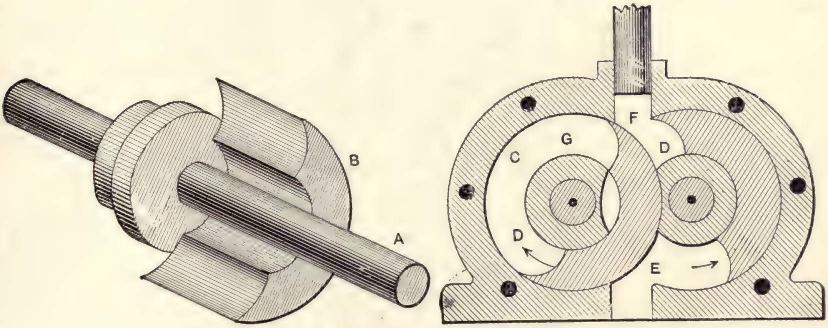


FIG. 101.—Behren's Rotary Pump.

being forced out of the pump. Water under pressure in *F* cannot pass the surfaces of the pistons, which are always in contact with the fixed surfaces of *C* or *D*. It is seen in the figure that after a little motion, the right-hand piston touches the surface on the left-hand liner *D*, and then it acts as an abutment while the water in the space *G* is discharged. One piston must come in contact with the liner of the opposite side before the other piston breaks contact with its opposite liner. In this manner there is always a barrier for a free passage from the discharge to the suction. This was improved in the Portland rotary pump of 1882, when the contact with the central abut-

ment made a line contact on the center line, making it unnecessary to have stationary sleeves, hence that part was eliminated.

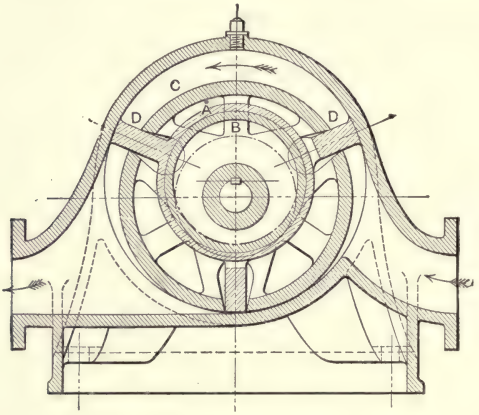


FIG. 102.—MacFarland's Rotary Pump.

The rotary pump of MacFarland, 1875 (Fig. 102), represents a development of the Trotter type of rotary pump. The construction of this pump as well as its action can be seen from the

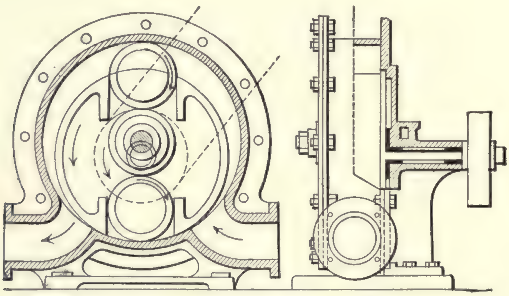


FIG. 103.—Phillip's Rotary Pump.

figure. The ring *A*, carrying the piston blades *D*, is supported on a stationary ring *B* projecting from the back head of the pump. The driving ring *C* is carried from the driving shaft, which is not placed at the center of the cylindrical barrel of the pump. The action of the blades on the water is clearly seen.

A pump somewhat similar to this was brought out by Phillips about ten years later (Fig. 103). In this pump the vanes attached to a sliding rotating cylinder were replaced by a cylindrical roller. This arrangement would eliminate much of the friction, although, at the center and periphery, there is still considerable friction produced by the slipping of the rollers against the sides of the driving slot.

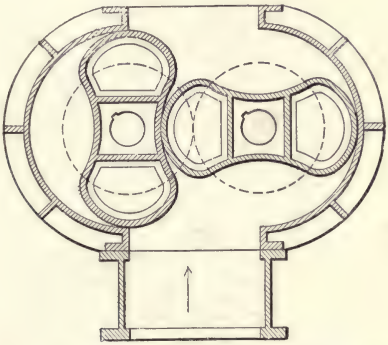


FIG. 104.—Wilkin's Two-Lobed Rotary Pump.

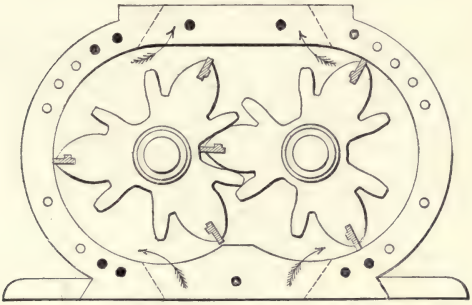


FIG. 105.—Silsby Rotary Pump.

The pump (Fig. 104) of John T. Wilkin, designed about the year 1892, consists of two similar rotating lobes each having two hypocycloids and two epicycloids. These two curves will work together with uniform velocity ratio. There is sliding contact with the cylinder walls and at this point there may be considerable leakage.

The use of the curves employed for gear-tooth profiles is not uncommon, as may be seen in Fig. 105, which represents the form of pump used on the Silsby fire engine, and in Fig. 106,

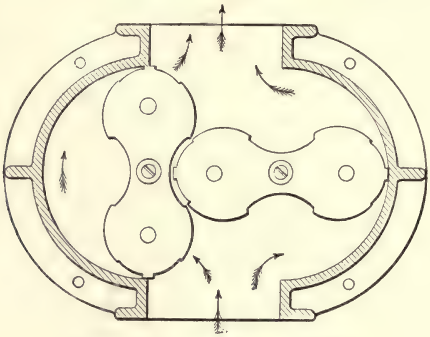


FIG. 106.—Root Rotary Pump.

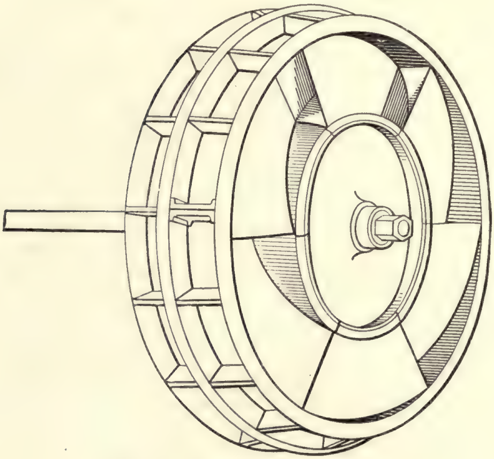


FIG. 107.—Allis Screw Pump.

which represents the form of propeller used by the Root Company. These last three pumps are modifications of the early form of Servière, illustrated in Chapter I. In the Root form the number of teeth is reduced to two for each wheel.

A development of the early form of **screw pump,** Chapter I, is shown in Fig. 107. This pump is for low lifts, being used in

this country when water is to be raised a few feet into a stream, there to mix with the sewage of a town. The screw acts on the water, forcing against a static head. For low lifts these

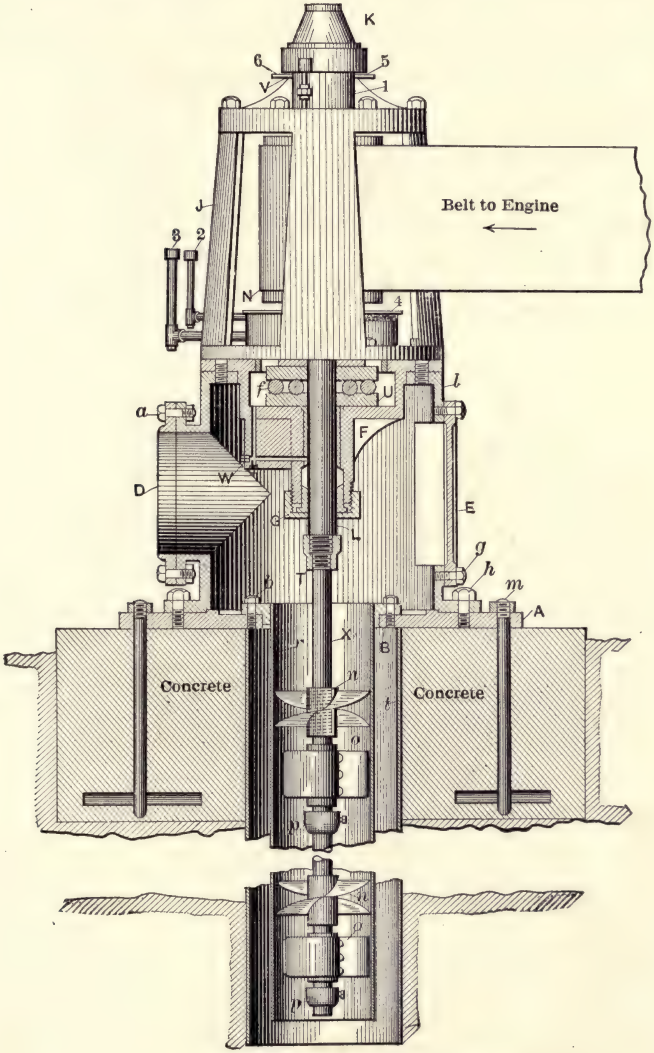


FIG. 108.—Wood Propeller Pump.

wheels, as well as the scoop wheels, are quite effective. A development of the same idea is shown in the Wood propeller pump

(Fig. 108). In this pump a vertical shaft guided by bearings *OO* carries a series of helical surfaces or screws *NN*, which fit close to the side of the pump pipe. Rotating this by means of a belt, as shown in the picture, the water is forced upward. An electric motor may be used for driving.

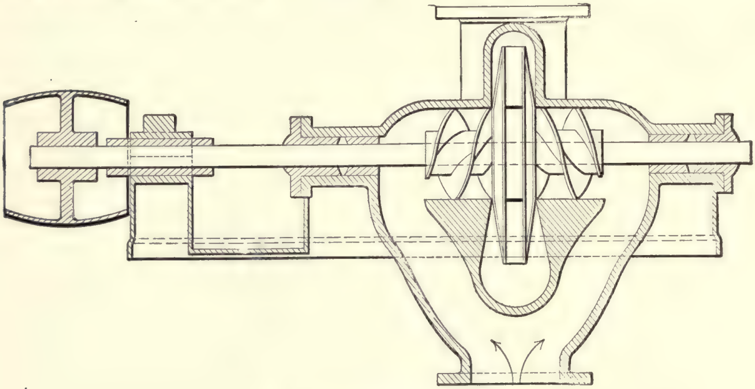


Fig. 109.—Helicoidal Pump.

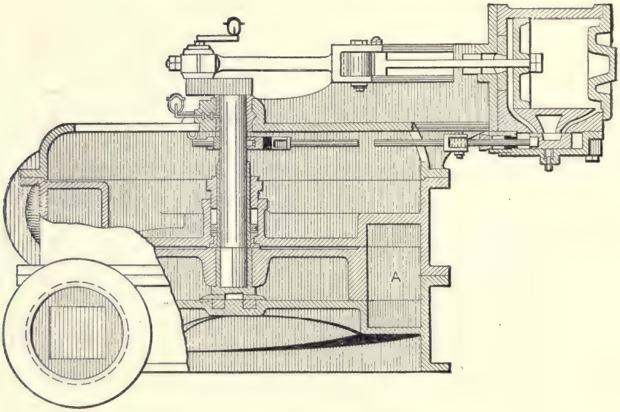


Fig. 110.—Bolton and Imray Helical Pump.

Wade & Cherry's **helicoidal pump** (Fig. 109) is a combination of the screw pump and the centrifugal pump. This was brought out in 1886. It is so arranged that the water is first forced toward the centrifugal impellers by the screws, after which it is discharged into the central wheel,

from which place it passes into the volute chamber around the wheel.

The Boulton & Imray **helical pump** (Fig. 110) was described in 1872. It is an impeller pump. The casing contains a helical passage through which the water goes from one level to another. The height of the passage is about twice its width and two-thirds the pitch of the helix. In this manner it is possible to place a series of square paddles *A* on a wheel rim of such a width that they cut off one-half the height of the passage, as seen in the figure. The action of the paddles is to

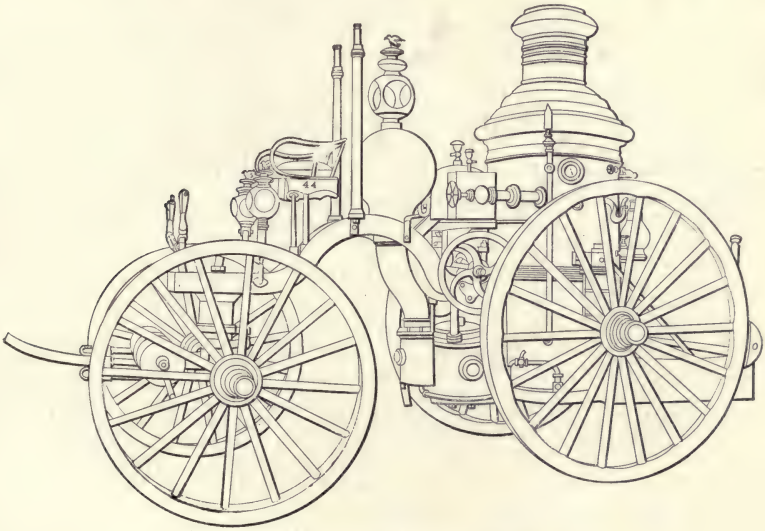


FIG. 111.—Gould Fire Engine.

impel the water through the casing. The pump shown in the figure had a wheel 3 feet 6 inches in diameter outside of the blades, the blades themselves being 6 inches square. The cylinder of the driving engine was 10×8 inches. Originally of the form shown it was changed in a later design by the use of the three-cylinder Brotherhood engine.

The steam fire pump brought out in the last decade of the first period considered in this work was improved during the following years, and by the year 1876 a number of them were in use.

The Gould steam fire pump (Fig. 111) exhibited at the Centennial Exposition in 1876 illustrates to what degree the steam fire engine had developed from the early form of Braithwaite and Ericsson described in the last chapter. The boiler has now been placed in a vertical position and the exhaust steam has been used to produce a draft in place of the early bellows. The engine has been made vertical also and a fly wheel has been added to make the operation of the pump more regular. This type of engine was likewise manufactured by Ahrens and the firm of Clapp & Jones in America. Fig.

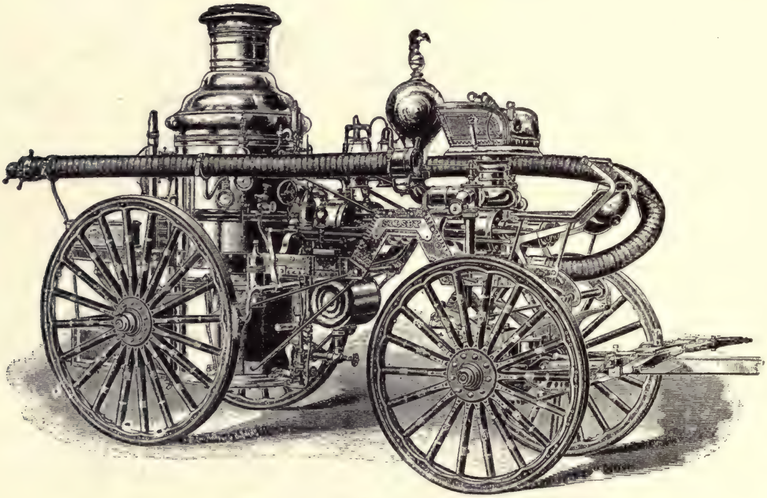


Fig. 112.—Silsby Fire Engine.

112 illustrates the application of the rotary pump and engine for fire service. The Silsby engine is one found in many of our large cities, and although there is considerable slip in the pump, its light weight is an important factor. These pumps were introduced about the middle of the last century. The rotary steam engine is seen near the boiler while the rotary pump appears beneath the driver's seat. The pump was of the type shown in Fig. 105, the steam engine being quite similar.

The use of horizontal engines is also found in the types of American fire engines, as may be seen in Fig. 113. The Button

engine is the product of one of the oldest builders of fire engines and the type is of value in that the parts are less complex than those in which the fly wheel is used. The early hand pumps

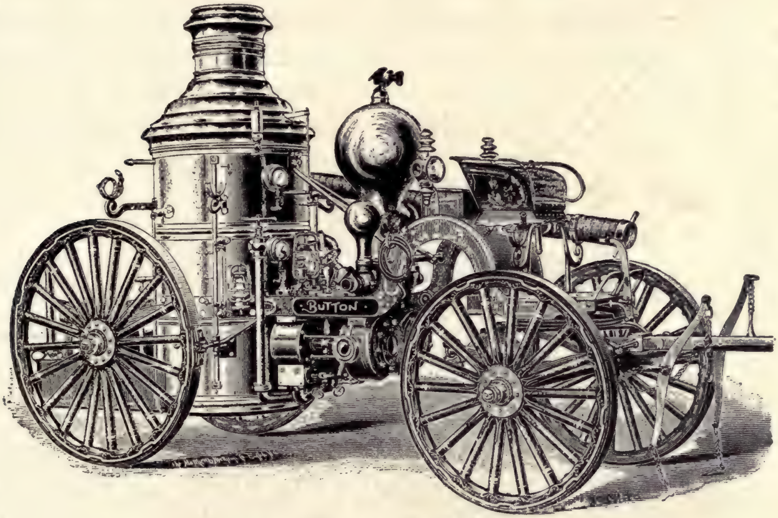


FIG. 113.—Button Fire Engine.

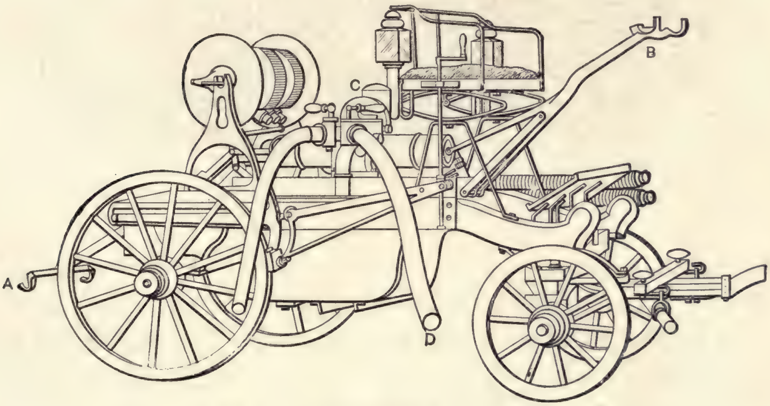


FIG. 114.—Hand Fire Pump of 1873.

were continued in many places even at this late date. Fig. 114 illustrates a pump exhibited at the Vienna Exposition of 1873. This machine was hauled by horses, but it was operated

by men. The handles *A* and *B* are attached to the pump plungers. The suction is connected to the pipe or hose *C* on the rear of the pump, not clearly shown in the figure, while the discharge is delivered through *D*.

Fig. 115 illustrates a modern type of fire engine of recent design. The pump is vertical with the steam cylinder near the boiler; the pump end near the point of attachment to the fire hydrant, air chambers on the suction and discharge sides of the pumps, valve boxes in accessible places and all parts open. The object sought in the modern fire engine is a machine always

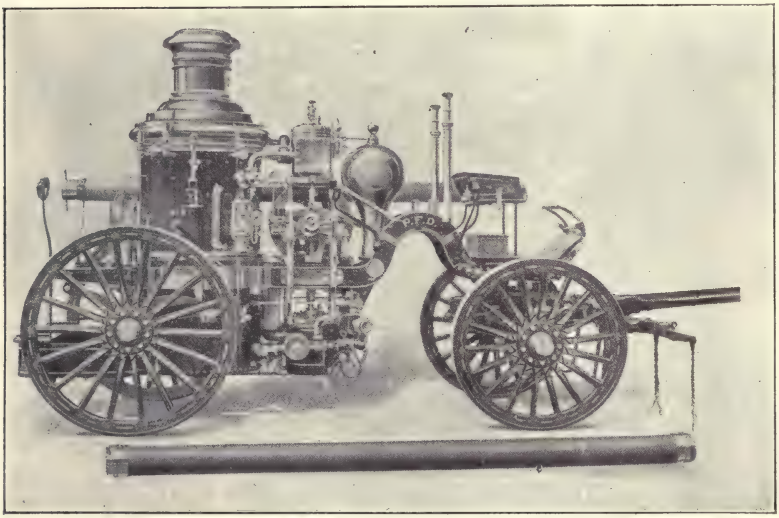


Fig. 115.—Modern Fire Engine, American-La France Co.

ready for a definite, positive service in which all parts are accessible for operation, maintenance, and repair.

The **air lift pump** was patented by James B. Frizell in 1880, and Julius I. Pohle made a number of improvements covered by patents granted in 1886. The pump (Fig. 116) consists of an air pipe *A* which dips into a well *B* and passes below the lower end of a larger pipe *C*. Compressed air is driven through the pipe *A* and discharges against the water pressure due to the depth of immersion of the discharge nozzle. The air escapes

from the nozzle and acts on the water above considered as a piston, or the air passes up through the water, which is thus made lighter and is forced upward by the static pressure of the water in the well. The figure shows beads of water acting as pistons with air between, expanding as the head is reduced by the upward motion of the water. This is probably the

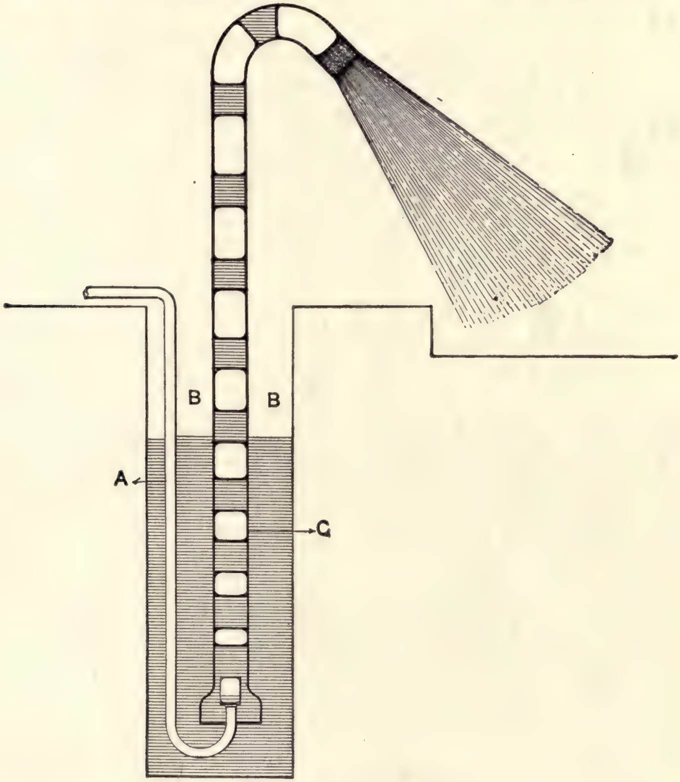


FIG. 116.—Air-Lift Pump.

manner of action of the pump after is it in operation, although in starting it is likely that the reduction of density by aëration causes the first flow. Although this form of pumping apparatus was not extensively used or seriously thought of before the work of Pohle in 1886, the method had been proposed earlier. Collom's Lectures on Mining for 1876, delivered in Paris, described this, and Gerlach in a paper points out that Loescher

of Freiberg described a similar apparatus in a pamphlet printed in 1797.

Another form of pumping apparatus using air is that invented by Professor Elmo G. Harris about 1900. In this pump (Fig. 117) air is compressed by the compressor *A* into a pipe

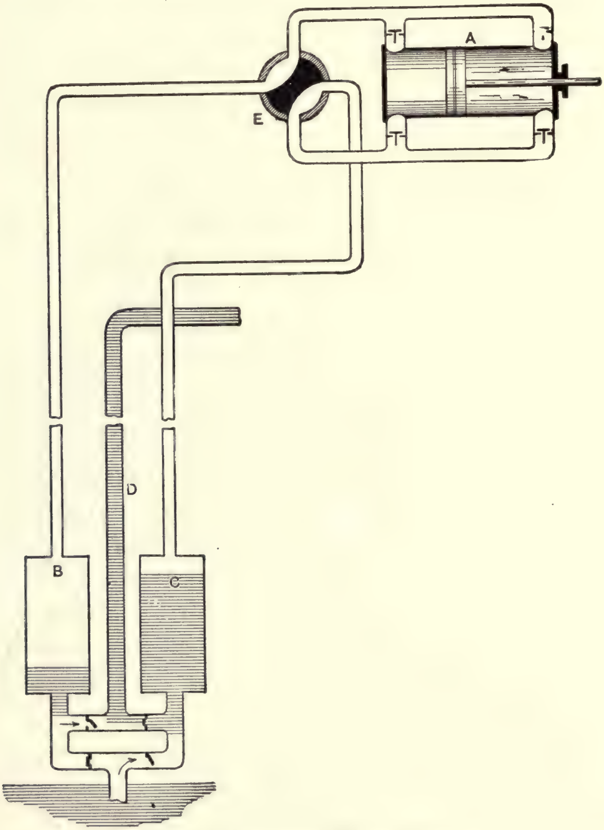


Fig. 117.—Harris Lift-Air Pump.

leading to a tank *B*. This compressed air acts on top of the water contained in *B* and drives it out of the discharge pipe *D*. The suction side of the compressor is connected with a pipe leading to the tank *C* and the reduction of pressure within the tank draws water into *C*. At the proper time the valve *E* is shifted, changing the suction and pressure sides so that water

is sucked into *B* and driven from *C*. There is an equalization of pressure through the valves of the compressor as soon as the shifting valve *E* is turned and before the compressor starts to drive out the water from *C*. This utilizes part of the energy used in compressing the air into cylinder *B*.

These two air lift pumps of Pohle and Harris have the advantage that a number of them may be driven from a central plant and there is little loss from such an arrangement. The air lift pump raises a larger quantity in a given time than would be possible with a deep well pump. The Harris pump may be installed in places where other pumps are impossible,

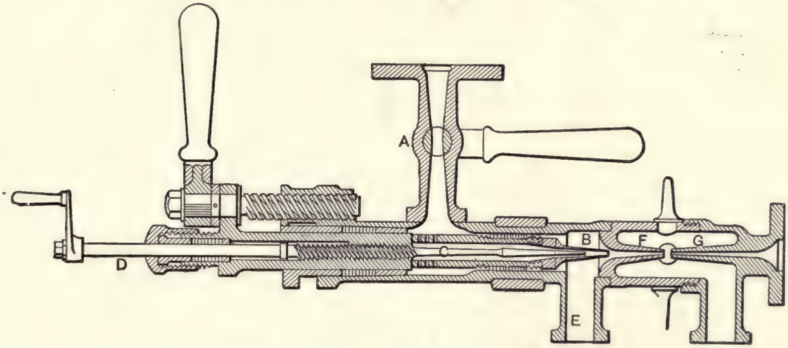


FIG. 118.—Giffard Injector.

and forms a positive type of pump. These pumps have specific advantages which will be pointed out when their design is considered.

In 1858 Henri Jacques Giffard patented his invention of the steam injector—one of the simplest devices for pumping water into boilers. In the earliest form (Fig. 118) steam entered through the cock *A* and passed through a series of small holes into the interior of a tube having a nozzle *B* at its extremity. The core or rod *C* controlled by *D* closes the nozzle or regulates the amount of opening. The steam acquires a high velocity in the nozzle, causing a vacuum in the space *E*, which raises water to that point, where it mixes with the steam. As the steam condenses and imparts a high velocity to the water, the cross-section of the mixture grows smaller and

consequently the combining tube F is made convergent. To change this high velocity into pressure, Giffard followed the converging combining tube by a diverging delivery tube G. In this manner the velocity was reduced and the pressure was so increased that the water could enter against the pressure of the boiler.

The idea of using the power of a moving jet was not new. According to Kneass a crude injecting apparatus was used as early as 1570 by Vitrio and Philibert de Lorme. In 1818 Mannoury d'Ectot patented a device which was quite similar to the injector, and Bourdon in 1857 patented a combination of convergent and divergent tubes for transforming the energy of a moving jet.

The latter invention resembled Giffard's, but the writings of the inventor of the injector, made seven years before Bourdon's invention, in which he explained the theory of the injector, proved that he should receive the credit for this invention.

The manufacture of these machines was at once undertaken in various countries by licensees. In America William Sellers & Co. began their manufacture, while Sharp, Stewart & Co., made them in England.

The Giffard design was an excellent one. Many ideas were suggested in his patent specifications and pamphlets, so that although changes were made in the injector as manufacturing processes were developed, and as more knowledge was had of the properties of jets and steam, the original ideas are still to be seen in the present-day injector.

The operation and design of injectors will be considered later with the modern form of injector, yet it is well to mention the names of some of those who added to the development of the injector. They are: Millholland, Rue, Sellers, Hancock, Williams, Loftus, Bancroft, and Kneass in America; Robinson & Gresham, Turck, Hamer, Metcalf, and Davies in England; Haswell, Pradel & Krauss and Körting in Germany; Cuau, Bouvret, Polonceau, and Delpeche in France.

A pump in which the driving steam touches the water is known as the **pulsometer** (Fig. 119). This is really a develop-

ment of the old Savery engine. Steam is admitted at *A* and if the ball is resting over the left-hand opening steam will enter *C* and drive out the water. A sudden change in the steam pressure when the water surface reaches the level of the discharge valve causes the ball to close the right-hand side, and then the condensation of the steam on that side draws water through the suction *D* and suction valve *E*, as shown in the figure.

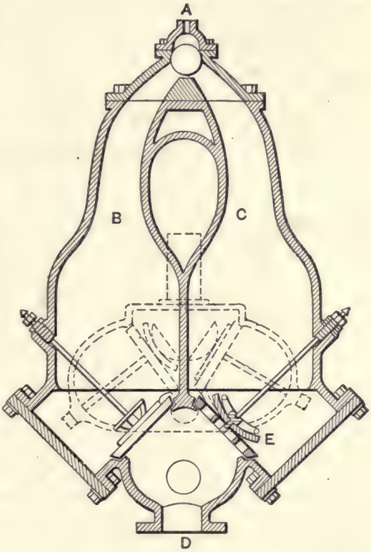


FIG. 119.—Pulsometer.

When the water is driven from *B* the ball is forced over to the left and the operation is repeated.

This type of pump is easily applied, and in contracting work it is used for this reason, although it is most expensive when its thermal efficiency is considered. For this kind of rough and rapid work it may not cost more in money to operate this than other forms of pumps.

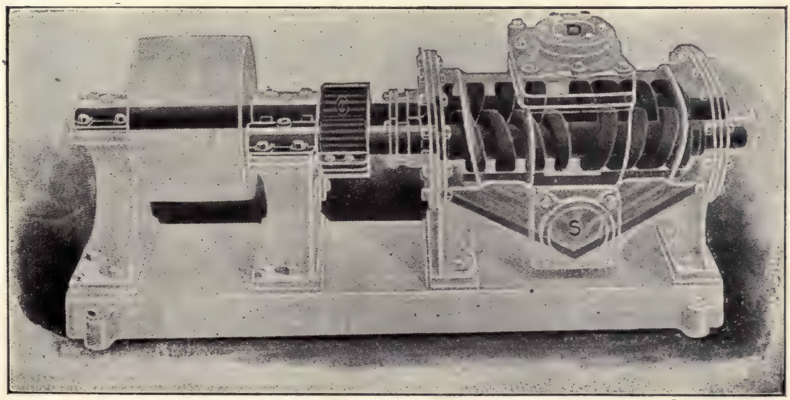


FIG. 120.—Quimbly Screw Pump.

The Quimby pump (Fig. 120) is a development of the pump shown in Chapter I, as the invention of Rèvillion. Its operation is evident from the figure, which shows right- and left-handed screws meshing together so as to form abutments for each other, catching the liquid in the threads, thus forcing it onward.

The latest form of pumping apparatus designed is that of Herbert A. Humphrey, described by him in *Engineering*, November 26, 1909. It is a combined gas engine and pump. The pump (Fig. 121) consists of an explosion head *A*, a suction

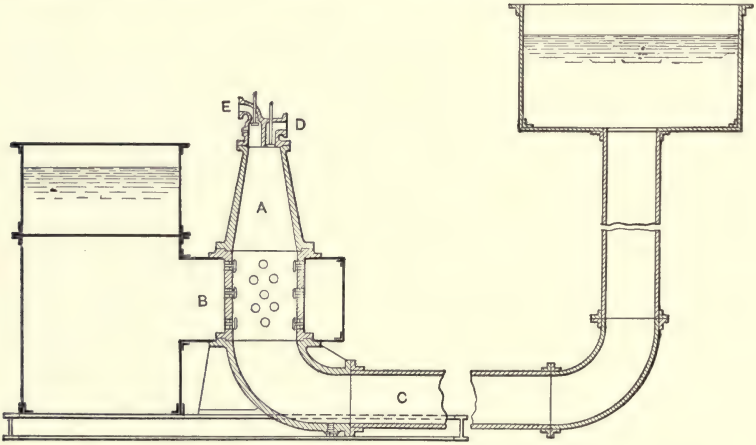


FIG. 121.—Explosion Pump of Humphrey.

portion *B*, a discharge pipe *C*. Assume that a compressed charge of gas and air is exploded in the head *A*. The force of this combustion drives the water from *C* into the reservoir, and the energy given to the water by the excess pressure in *A* is dissipated only after the pressure in *A* has been reduced to a vacuum. When the pressure at *B* within the pipe system is reduced sufficiently, water is drawn in through the suction valves. After the water is brought to rest in *C*, the gases in *A* are compressed by the static pressure from the reservoir, which forces the water backward, developing a certain amount of velocity. During this operation, however, the valve *D* is opened automatically, and the burned gases are exhausted.

When the water reaches a point near the valve *D* the valve is closed, and before the velocity which has been set up in *C* can be dissipated the water and gas in *A* are compressed beyond the pressure corresponding to the static head. This means that there is another surge toward the reservoir, followed by a vacuum in *A*, which is destroyed by opening the air and gas valve *E*, permitting a fresh charge to enter. This suction stroke is followed by another backward surge, during which the explosive mixture in *A* is compressed. At the proper time this is exploded by an electric spark, and the operation is repeated. The valves *E* and *D* and the ignition apparatus are operated by the surging water in the system. The device is so arranged that *E* opens on an expansion stroke after *D* closes at the end of a compression stroke.

Professor W. C. Unwin made a test on this pump in connection with a gas producer, developing 16 pump horse power on 1.063 pounds coal per delivered horse power hour or 12,243 B.T.U. per P.H.P. hour. The data for this test are given in the table below:

Lift.	P.H.P.	Cu. ft. at 14.7–32° F.	Calorific Value.	B.T.U. per P.H.P. hr.	Lbs. Anth. per P.H.P. hr.
32.87	16.15	83.12	147.3	12,243	1.063
25.95	12.32	90.93	143.5	13,037	1.132
20.73	10.99	93.61	145.3	13,596	1.180

Mr. Humphrey does not claim to be the originator of a gas-driven pump, as he says this matter dates back to 1868, but he deserves credit for having built so simple a device which yet gives efficiencies higher than those of the most improved steam pumps.

CHAPTER III

MODERN FORMS OF PUMPS

In considering the actual pumps in use it is advisable to classify them in several ways: (*a*) In regard to the form of water displacer; (*b*) in regard to the number of displacements; (*c*) in regard to the method of operation; (*d*) in regard to the manner of packing; (*e*) in regard to the direction of the axis of the pump cylinder; (*f*) in regard to the arrangement of cylinders; (*g*) in regard to the use of the pump.

The general forms of that part of a pump used to displace water are the plunger, the piston, and the bucket, although such means as air, gas, and steam are used, as was shown in Chapters I and II. The plunger is usually cylindrical in section and forces the water of a pump by entering the space occupied by the water. Fig. 122 shows the construction of a simple plunger pump. This plunger enters the cavity filled with water and displaces an amount equal to the increase of volume of the plunger protruding into the cylinder.

The piston (Fig. 123) consists of a movable diaphragm tightly fitting against the sides of a cylinder, and forcing the water before it. This piston forces an amount of water equal to the area of the piston multiplied by the stroke.

The bucket is a piston containing a number of valve-closed passages through which the fluid may pass at the proper time. Fig. 124 shows the construction of a bucket.

The use of any one of these devices on a pump gives its name to the pump. Thus the names *plunger pump*, *piston pump*, or *bucket pump* designate a pump in which one of these displacers is used.

Single plungers and buckets, as ordinarily constructed, can lift water on only one stroke, while the piston displaces water

on both strokes. This gives rise to the classes, single- and double-acting pumps. Of course two plungers may be united

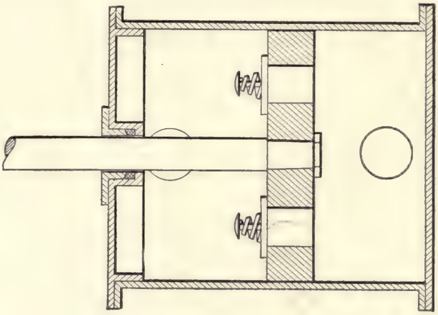


FIG. 124.—Bucket Pump.

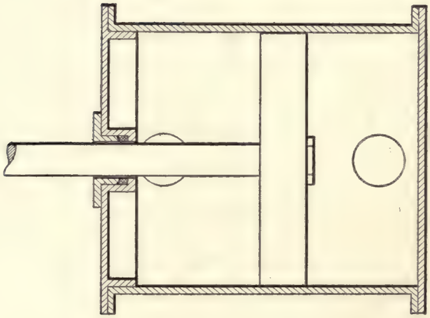


FIG. 123.—Piston Pump.

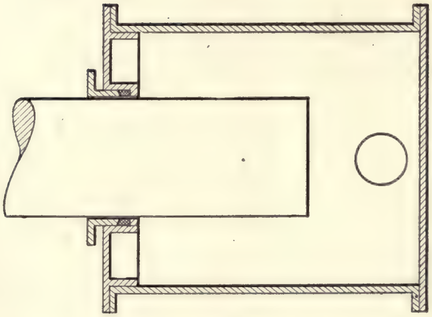


FIG. 122.—Plunger Pump.

by an outside or an inside connection and form a double-acting plunger pump as shown in Fig. 125, while a bucket or plunger pump may be so arranged that although it lifts only on one

stroke of every two strokes, it discharges on each stroke. Such pumps are known as differential pumps. Fig. 126 shows the arrangement for a plunger, and the same could be used with a bucket (Fig. 127). The piston rod is these cases is so enlarged that it is practically another plunger of about one-half the area

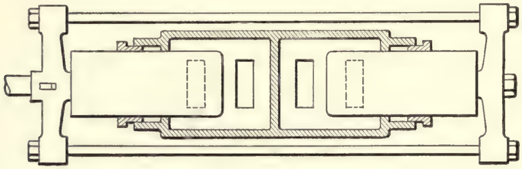


FIG. 125.—Double-Acting Plunger Pump.

of the main plunger or bucket. In this manner by connecting the lifting side of the pump to the discharge main through the other end of the pump, one-half of the water will be retained on this side during the discharge stroke of the main plunger, while one-half of the water is discharged. The portion of the

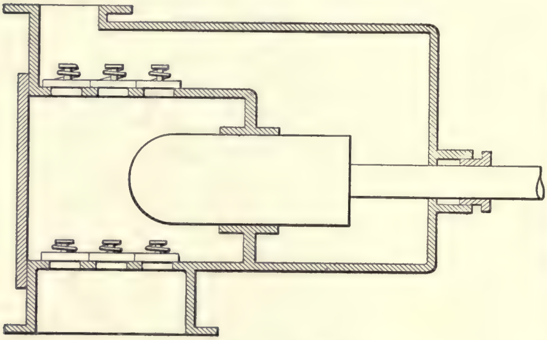


FIG. 126.—Differential Plunger Pump.

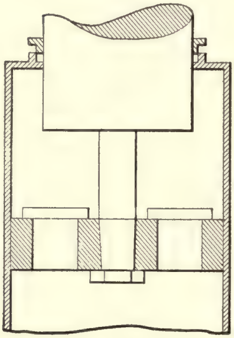


FIG. 127.—Differential Bucket Pump.

water retained is delivered on the suction stroke of the main plunger or bucket. In this manner the discharge is made more regular, although the suction occurs on every other stroke.

The method of operation gives another classification. *Steam pumps* are those driven by steam pistons, while *compressed-air pumps* are those in which air is used in place of

steam. Where the steam and water pistons are directly connected these are known as direct acting, although if a fly wheel is connected to the system they are known as *fly-wheel pumps.*

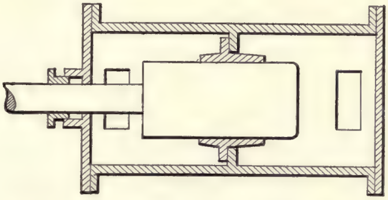


FIG. 128.—Plunger and Ring Packing.

Power pumps are those driven by belts or gearing, and *electric pumps* are those directly driven by electric motors. The use of air or gas in direct contact with the water has given the class of *air-lift* and *gas-driven pumps* described in Chapter II,

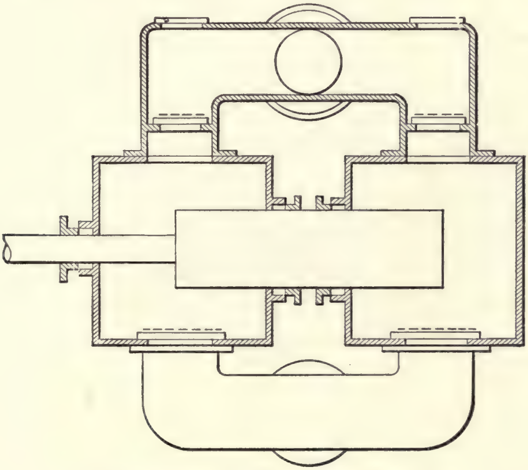


FIG. 129.—Central Outside Packing.

while in the injector and pulsometer the water is moved by the direct action of steam.

The classification due to the method of packing the water piston or plunger divides pumps into *outside-packed pumps* (Fig. 125), and *inside-packed pumps* (Fig. 128), as well as central-

and end-packed machines. A special type of inside packing is known as the plunger and ring type. Fig. 128 shows the arrangement of this type of packing. The plunger in this case passes through a long sleeve or ring in which the resistance against the flow of water is so great that there is little or no leakage. Fig. 125 shows an end packing while Fig. 129 shows central packing.

Vertical and horizontal pumps are distinguished by the direction of the axis of the pump.

A pump with a single water cylinder is called a *simplex pump*, while two-cylinder pumps are called *duplex*, and three-cylinder, *triplex*. The first two names, however, are usually associated with direct-acting pumps of small or medium size, while the last term is usually applied to a type of single-acting power pump in which there are three cylinders.

The last classification of pumps is that due to the use of the pump. It is one of the largest classifications and will demand a more detailed consideration. In examining these different classes the peculiar features of each will be pointed out. For those types which will not be considered later in this work, a full description will be given here.

Boiler-Feed Pumps. This type of pump is, in all probability, the most common. The conditions under which such pumps are used determine many of their details. In general the pump has a water piston or plunger which is about one-half the size of the steam cylinder. The object of such a difference is to have ample fcrce with any amount of steam pressure to drive water into the boiler against the steam pressure which operates the pump. The pump must be as simple as possible, since it is to be handled by unskilled men. It must have few parts which are liable to breakage or disarrangement. As the variation in pressure in the discharge line is not important and as the pumps are rarely run at full speed, it is quite customary to install these pumps without air chambers on the discharge or suction.

These pumps are either simplex or duplex, as may be seen from Fig. 130 and Fig. 131. Fig. 130 shows two small Worth-

ington pumps, the upper one being a piston pump and the lower an end outside-packed plunger pump. These are both of the same size, 6 inches diameter of steam cylinder, 4 inches diameter of water cylinder, and a common stroke of 6 inches. This is usually written as $6 \times 4 \times 6$ inches. Fig. 131 shows a

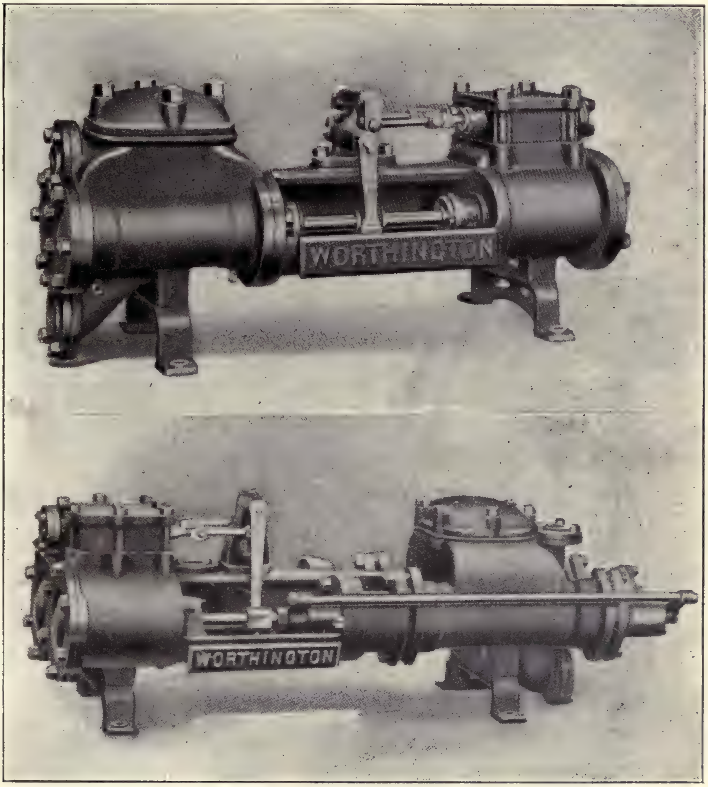


FIG. 130.—Duplex Boiler Feed Pumps.
(Sizes $6 \times 4 \times 6$.)

$10 \times 6 \times 12$ inch Knowles simplex boiler-feed pump. The water end of this pump is of the piston type.

For heavier water pressures the boiler-feed pump is made with steam cylinders, large when compared with the water end, and the valves are placed above the cylinder casting in separate valve pots as shown in Fig. 132. This pump, 12 inches

and $17 \times 10 \times 15$ inches, is a compound pump in which two steam cylinders are used on each side of the duplex pump for

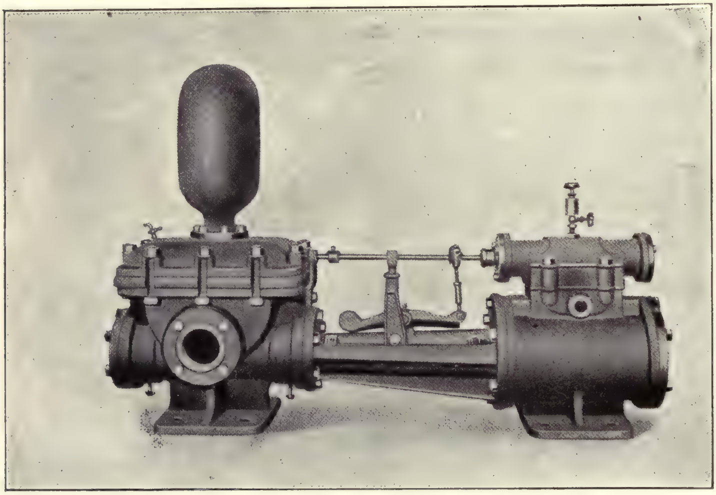


FIG. 131.—Simplex Boiler Feed Pump. (Size $10 \times 6 \times 12$.)

the purpose of getting greater economy. Fig. 133 shows a different design for the same type of pump. The outside

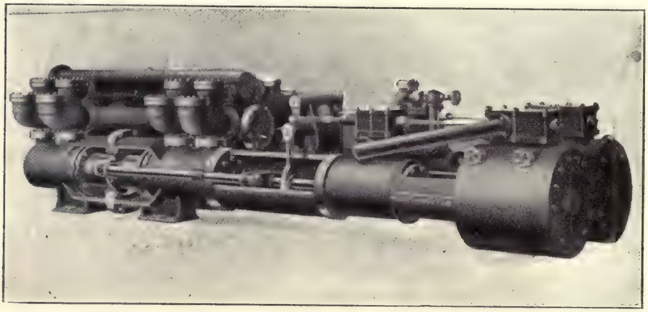


FIG. 132.—Compound Outside Center Packer Boiler Feed Pump. (Size 12 and $17 \times 10 \times 15$.)

packing in this case is of the end type instead of the center type shown in Fig. 132.

For use in marine installations where floor space is valuable

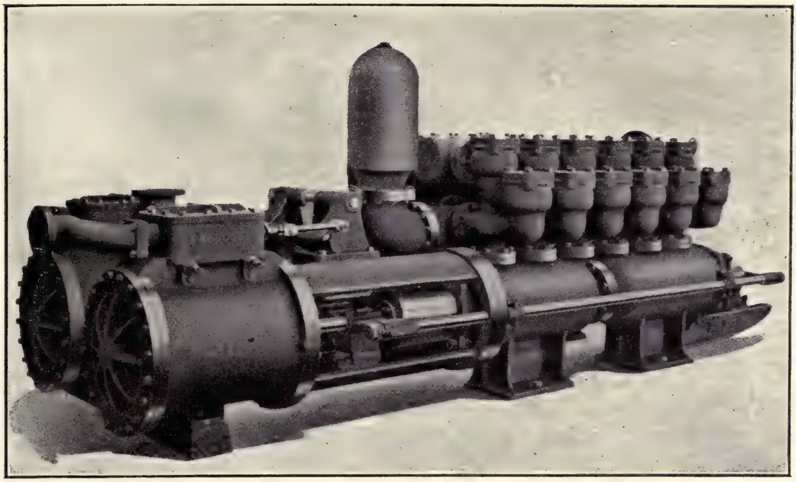


FIG. 133.—Outside Packed Plunger Boiler Feed Pump.

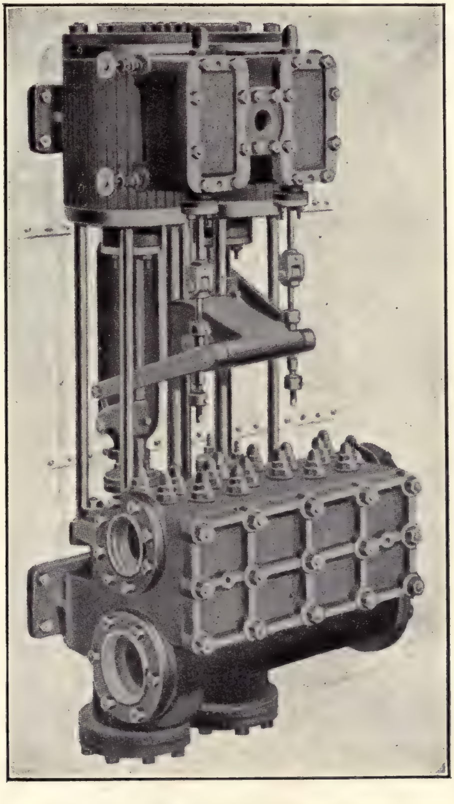


FIG. 134.—Marine Boiler Feed Pump.
(Size $10 \times 7 \times 12$.)

vertical pumps are designed. These are often known as *Admiralty Pumps.* Fig. 134 shows a Davidson duplex piston pump bolted to a bulkhead of a vessel. It is to be noted that this pump has its valves so placed that they may be examined readily. The suction and discharge pipes may be attached to either side, and caps on the cylinder heads allow an examination of the water piston.

To give some idea of sizes of boiler-feed pumps the following tables have been taken from catalogues of pump makers:

THE WORTHINGTON BOILER-FEED PUMP

Pressure Pattern—For 250 Pounds Pressure.

These pumps have four single-acting, outside-packed water plungers, working through adjustable stuffing boxes in the ends of the water cylinders. The valves are of brass, guided from below by wings and controlled by composition springs, and are located in separate valve chambers or pots designed to withstand the heavy pressures to which this pump may be subjected.

Diameter of Steam Cylinders.	Diameter of Water Plungers.	Length of Stroke.	H.P. Boiler, Based on 45 lbs. Water per hour, which Pump will Supply at Slow Speed.	Sizes of Pipes for Short Lengths to be Increased as Length Increases.				Approximate Space Occupied, Feet and Inches.	
				Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Length.	Width.
4½	2	4	70	½	¾	1½	1	3 9½	1 3
5¼	3	5	190	¾	1¼	2½	2	4 11	1 6½
6	3½	6	290	1	1¼	2½	2	5 7	1 6½
7½	4½	6	470	1½	2	4	3	5 10	2 1
7½	4½	10	670	1½	2	4	3	8 3	2 1
9	5	10	800	2	2½	4	3	8 3	2 1
10	6	10	1200	2	2½	6	5	8 10	4 2
12	7½	10	1400	2½	3	6	5	9 11	4 3
[1] 14	8½	10	1800	2½	3	8	7	10 1	3 3
12	7½	15	2000	2½	3	6	5	11 9	4 3
[1] 14	8½	15	2700	2½	3	8	7	11 2	4 0
[1] 17	10	15	3700	2½	3½	8	7	12 2	4 3
[1] 20	12	15	5200	4	5	10	8	14 5	4 5

[1] These sizes have four center-packed plungers.

PRESCOTT DUPLEX OUTSIDE-PACKED PLUNGER POT-FORM BOILER-FEED PUMPS

For 300 Pounds Working Pressure.

The water ends are of the "pot form," and have four single-acting outside-packed water plungers. The water valves are designed for either hot or cold water and arranged so as to be readily accessible. All water passages are large and direct.

The plunger stuffing boxes are very deep and fitted at the bottom with a removable brass ring which can be replaced when worn.

These pumps are especially adapted for boiler-feeding service in electric lighting and railway stations or in other plants where high pressures are carried.

Size.			H. P. Boiler, Pump will feed, based on 30 lbs. per H. P. Hour.	Diameter of Pipe Openings.				Space Occupied, Feet and Inches.	
Steam Cylinder.	Water Plunger.	Stroke.		Steam.	Exhaust.	Suction.	Discharge.	Length.	Width.
8	4	12	500	1½	2	3½	3	9 9¼	4 6½
10	5	12	800	2	2½	4	3½	10 2½	4 6½
12	6	12	1,200	2	3	6	5	10 3¼	5 9
12	7	12	2,000	2	3	6	5	11 4	5 5
14	8	18	4,300	2½	3½	7	6	14 3	4 3
16	10	18	6,850	2½	3½	8	6	14 5½	4 1½
18	10	18	6,850	3	4	8	6	14 7½	4 1½
24	14	24	25,000	5	6	12	10	19 0¾	5 11¾

THE DAVIDSON VERTICAL DUPLEX PUMP

For a Pressure of 250 Pounds.

In the Davidson vertical duplex pump there are but few joints, all of which are visible and easily renewed, the water valves are easily accessible for examination or renewal, the water pistons can be packed from the upper end of cylinder and are fitted with fibrous or metallic packing. Water ends of cast iron are composition lined and fitted, or entirely of composition when specially ordered.

Steam Cylinder.	Water Cylinder.	Stroke, Inches.	Gallons per Single Stroke of each Piston.	H. P. Boiler, based on 30 lbs. of Water per H. P. per Hour, which the Pump will supply with Ease.	Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Discharge Pipe.
4	2½	4	.084	165	½	¾	2	1½
4½	2¾	6	.154	300	½	¾	2½	2
5½	3½	6	.25	500	1	1¼	3	2½
6	4	8	.435	870	1	1¼	3	2½
7	4	8	.435	870	1¼	1½	3	2½
7	4½	8	.55	1100	1¼	1½	4	3
8	5	10	.85	1700	1½	2	4	3½
8	5	12	1.02	2000	1½	2	4	3½
9	5½	10	1.03	2000	1½	2	4½	4
10	6	10	1.225	2450	2	2½	5	4½
10	6	12	1.469	2900	2	2½	5	4½
12	7	12	2.00	4000	2	2½	6	5
14	8	12	2.61	5000	2½	3	7	6
14	8½	12	2.94	6000	2½	3	7	6

Suction and discharge openings on both sides.

Capacities for boiler feeding are based on a speed of 60 single

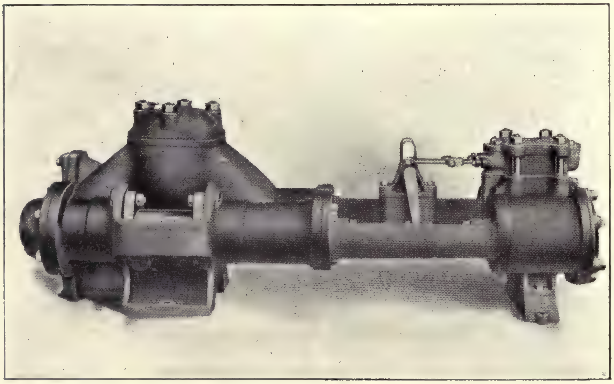


FIG. 135.—Worthington Packed-Plunger Pump.
(Size $7\frac{1}{2} \times 5 \times 6$.)

strokes a side per minute; for other services, pumps should be run at a piston speed of 30 to 80 feet a side per minute, and

in cases of emergency can be speeded up greatly in excess of this.

General Service Pumps are those which are intended for the pumping of water for various purposes; drainage, elevator work, small water supply or any other work of a general nature. These pumps are usually designed for specific pressures. For water pressures of 200, 250, and 300 pounds per square inch the outside packed plunger water ends (Fig. 135) are used, while

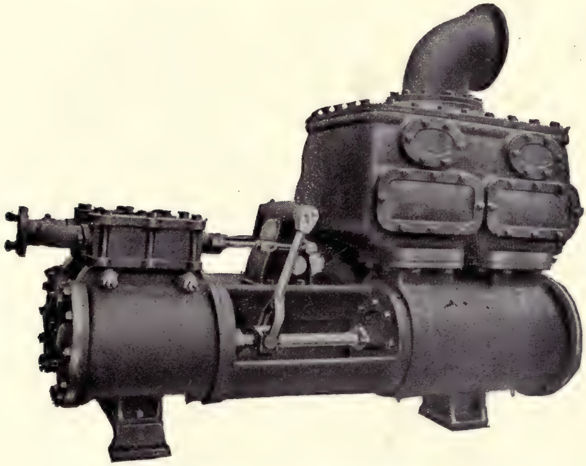


Fig. 136.—Tank Pump.

(Size 12×15×15.)

with pressures of about 150 pounds piston pumps are used. When the water pressure is from 35 to 50 pound the pump is known as a *tank pump*, and although the general form is the same as that of the other pumps, the parts are made lighter and the steam cylinder is made smaller than the water end. In all cases of these pumps the variation from one class to another depends on the pressure to be carried. Fig. 136 shows the type of tank pump, while the following table gives the sizes in use by one manufacturer.

THE WORTHINGTON PISTON PUMP

For Tank or Light Service.

Diameter of Steam Cylinders.	Diameter of Water Pistons.	Length of Stroke.	Gallons per Revolution.	Maximum Revolutions per Minute.	Maximum Gallons per Minute.	Sizes of Pipes for Short Lengths to be increased as Length increases.				Approximate Space Occupied, Feet and Inches.			
						Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Length.		Width.	
3	2¾	3	·3	80	24	⅜	½	1¼	1	2	10½	0	9¼
4½	3¾	4	·75	75	56	½	¾	2½	1½	2	11	1	1
5¼	4¾	5	1.51	70	106	¾	1¼	3	2	3	3	1	4
6	5¾	6	2.65	65	172	1	1¼	4	3	3	10	1	5
7½	5¾	6	2.65	65	172	1½	2	4	3	3	11	1	10
6	7½	6	4·54	65	295	1	1¼	6	5	3	9	1	9
7½	7½	6	4·54	65	295	1½	2	6	5	3	9	1	9
6	8½	6	5·84	65	380	1	1¼	6	5	3	9	1	10
7½	8½	6	5·84	65	380	1½	2	6	5	3	9	1	10
7½	6	10	4·75	54	256	1½	2	5	4	5	8	2	4
7½	7	10	6.52	54	352	1½	2	6	5	5	8½	2	4
10	7	10	6.52	54	352	2	2½	6	5	5	9	2	5
7½	8½	10	9.68	54	522	1½	2	6	5	6	0	2	4
9	8½	10	9.68	54	522	2	2½	6	5	6	0	2	4
7½	10¼	10	14.08	54	760	1½	2	10	8	6	4	2	8
9	10¼	10	14.08	54	760	2	2½	10	8	6	5	2	8
12	10¼	10	14.08	54	760	2½	3	10	8	6	9	2	8
12	12	10	19·37	54	1045	2½	3	10	8	6	5	3	0
12	14	10	26.44	54	1427	2½	3	12	10	7	0	3	4
12	15	10	30·38	54	1640	2½	3	12	10	7	0	3	4
14	15	10	30·38	54	1640	2½	3	12	10	7	0	3	4
12	14	15	39·70	40	1588	2½	3	12	10	7	10	3	10
12	15	15	45·61	40	1824	2½	3	12	10	7	10	3	10
14	15	15	45·61	40	1824	2½	3	12	10	7	10	3	10
17	15	15	45·61	40	1824	2½	3½	12	10	8	0	3	10
12	17	15	58.66	40	2346	2½	3	14	12	8	5	4	1
14	17	15	58.66	40	2346	2½	3	14	12	8	5	4	1
14	19	15	73·27	40	2931	2½	3	16	14	8	0	4	1
17	19	15	73·27	40	2931	2½	3½	16	14	8	1½	4	1
14	22	15	92.26	40	3940	2½	3	16	14	8	6	4	7

An additional charge is made for Tobin-bronze piston rods, brass water pistons, bed plates, or for any extras.

The water end is made of light construction for use on

services where the total water pressure to be pumped against is not over from 35 to 50 pounds per square inch. The ratios of steam cylinders and water pistons are suitable for raising liquids to moderate heights with ordinary steam pressures. This design is intended for use at railway water stations, breweries, distilleries, gas and oil works, tanneries, bleacheries, refineries, etc. The valves in the liquid end are furnished of material suitable for the liquid to be pumped. The liquid ends up to the size 14×10 are designed for pressures up to 50 pounds per square inch and the larger ends for a pressure of 35 pounds per square inch.

A tank pump used for pumping water from the bottom of a vessel is known as a *bilge pump* or *ballast pump*. These pumps are usually placed in a vertical position owing to the lack of room. On account of the small head against which they work the steam cylinder is much smaller than the water cylinder. Fig. 137 shows the form of ballast pump built by Worthington and the covers over the valve chambers show how simple it is to care for the valves. This is an important matter in pumps for marine service where working space is limited.

Fire Pumps are those installed for fire protection. When these are built according to certain specifications adopted by the Underwriters Associations they are known as *underwriters' pumps*. Fig. 138 shows such a pump with ample dimensions on the steam end. As will be seen later these pumps have gauges on the steam and water ends, air chambers on the suction and discharge pipes, direct hose connections on the pumps, a relief valve on the discharge main, and a name plate with certain data in regard to the pump on the air chamber. The water parts are of bronze and there are other peculiar features which will be considered.

THE KNOWLES UNDERWRITER FIRE PUMPS

1904 Pattern

In these underwriter fire pumps the water passages, valve areas, and suction and discharge nozzles are all much larger

than in any ordinary pump, so a greater amount of water can be discharged without water hammer. Also, the steam and exhaust ports and nozzles are designed so as to give unrestricted passage to the steam. The pump is "rust proof,"

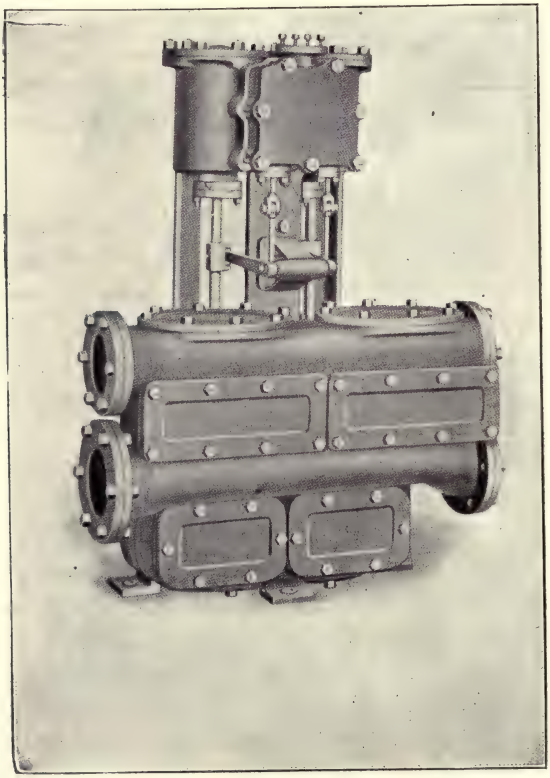


FIG. 137.—Worthington Ballast Pump.

(Size $7\frac{1}{2} \times 10\frac{1}{4} \times 10$.)

and will start instantly, after standing unused for a long time. The piston rods and valve rods are made from Tobin bronze. All stuffing boxes and glands are brass lined. Plungers and plunger sleeves are of composition, but the metals are differently mixed to prevent cutting. This mixture of the metals has been a subject of much experiment and study; it has now

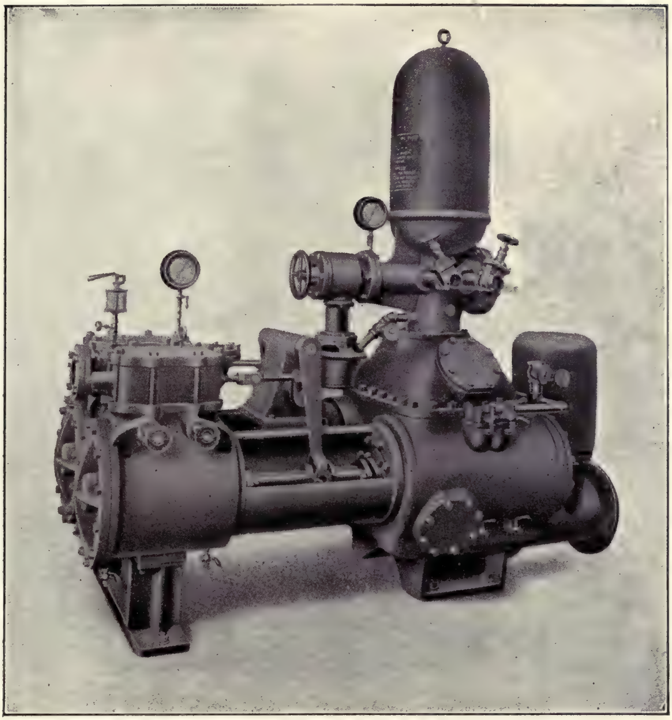


FIG. 138.—Knowles Underwriters' Pump.

(Size 18×10×12; 1000 gallons per minute.)

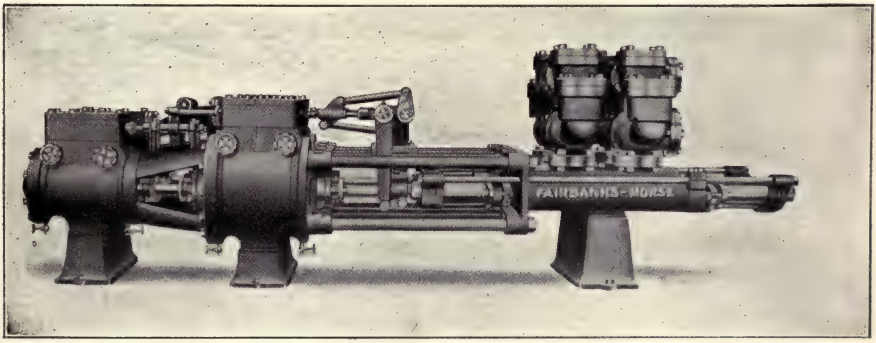


FIG. 139.—Compound Outside End-Packed Pressure Pump.

(Size 10×16×4×21.)

brought great success, as the parts work perfectly upon each other, without friction or impairment.

Each pump has the following fittings:

Capacity plate on discharge air chamber; stroke gauge, graduated on each end; vacuum, or suction air chamber; steam gauge, 5 inches diameter; water gauge, 5 inches diameter; relief valve of large capacity; relief valve discharge cone; set of brass priming pipes and special priming valves; required number of $2\frac{1}{2}$-inch Ludlow hose valves; one pint sight-feed cylinder lubricator; one pint hand oil pump, all according to the specifications.

On account of the larger passageways, brass parts and attachments mentioned, the pump necessarily costs more than an ordinary fire pump; but the cost by the gallon discharged is less, since the underwriter pump can deliver a greater quantity of water in the same length of time. It is also much heavier and stronger, with superior workmanship, and better protected from rust and accident.

Hose valves are not threaded, unless specially ordered, at slight extra charge. If these valves are to be threaded, a sample thread must be supplied by the purchaser, as there is no established standard, hose threads varying widely in different localities.

Diameter of Steam Cylinder, Inches.	Diameter of Water Plunger, Inches.	Length of Stroke, Inches.	Normal Capacities Gals. per Minute.	Number of Fire Streams.	Steam Pipe, Inches.	Exhaust Pipe, Inches.	Suction Pipe, Inches.	Discharge Pipe, Inches.
14	$7\frac{1}{4}$	12	500	2	3	4	8	6
16	9	12	750	3	$3\frac{1}{2}$	4	10	8
18	10	12	1000	4	4	5	12	8
20	12	16	1500	6	5	6	14	10

Underwriter fire pumps are built with compound steam ends, and power underwriter fire pumps are arranged to run with electric motors.

Pressure Pumps (Fig. 139) are those which are intended for hydraulic pressure installations in which pressures of 500 pounds and over are used to operate presses. Although these

pumps are sometimes of the fly-wheel tpye, the intermittent operation of the pump in many plants makes this type unsuitable, hence the direct-acting type is used very extensively. This pump is marked by small water cylinders, usually of the outside-packed plunger type. The valves are placed in valve boxes bolted to the water end on account of the complex casting which would result if these parts were integral portions of the water cylinders.

The Mine Pump (Fig. 140) is a form of pressure pump as it is used to pump water from great depths. The water end is

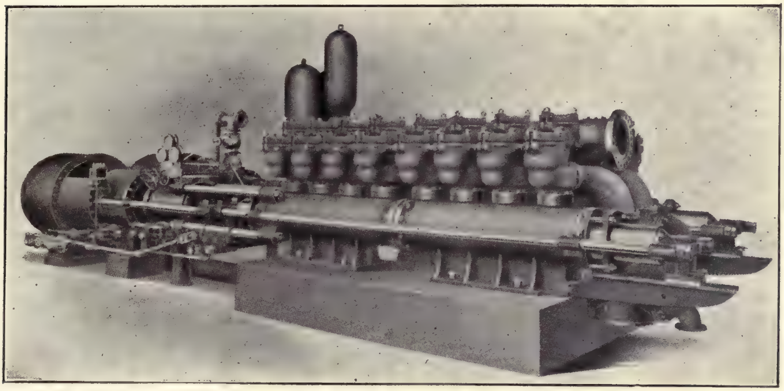


FIG. 140.—Compound Outside-Packed Mine Pump.

therefore quite similar in detail to the pressure pump. There are other reasons for the construction of the water ends in small parts. The action of the acid of mine waters is such that parts of the pump are rapidly corroded, hence the use of a number of small parts of similar form makes it possible to renew the part affected and not replace a whole end, as would be necessary if the water end was one complex casting. Moreover the number of different parts is reduced to a minimum by having most of the small parts similar for both sides of the pump. This cuts down the number of spare parts to be carried in the storeroom.

The table below gives the sizes used by the Worthington

Co. for their Lehigh pattern of pump used with pressures of 300 pounds per square inch and over.

THE WORTHINGTON MINE PUMP

Lehigh Pattern—For 300 Pounds.

The Worthington mine pump (Lehigh pattern) is specially designed to withstand the heavy pressures encountered in deep workings. These pumps can be arranged to operate either non-condensing or condensing, and are also made with compound and triple-expansion steam cylinders, where a saving of fuel is desirable. Pumps of this design of larger size or for heavier service can be furnished.

Diameter of Steam Cylinders.	Diameter of Water Plungers.	Length of Stroke.	Gallons per Revolution.	Revolutions per Minute.	Gallons per Minute.	Size of Pipes for Short Lengths to be Increased as Length Increases.				Approximate Space Occupied. Feet and Inches.	
						Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Length.	Width.
16	6	10	4.89	43	210	2½	3	6	5	9 11	5 2
18½	6	10	4.89	43	210	3	3½	6	5	10 0	5 2
16	7½	10	7.64	43	328	2½	3	6	5	9 11	5 2
18½	7½	10	7.64	43	328	3	3½	6	5	10 0	5 2
17	6	15	7.35	32	235	2½	3½	6	5	11 10	5 2
20	6	15	7.35	32	235	4	5	6	5	11 11	5 2
17	7½	15	11.48	32	367	2½	3½	6	5	11 10	5 2
20	7½	15	11.48	32	367	4	5	6	5	11 11	5 2
18	7	18	12.00	28	336	3	4	6	5		
22	7	18	12.00	28	336	4	6	6	5		
18	8	18	15.67	28	438	3	4	10	8		
22	8	18	15.67	28	438	4	6	10	8		

An additional charge is made when pumps are fitted with brass plungers and brass-bushed glands.

To designate the sizes, give the diameters of the steam cylinders and water plungers, and the length of stroke.

NOTE.—One revolution means four strokes, counting both sides.

When lower pressures are present and the water is not corrosive such a type shown in Fig. 141 may be used. The

pump is of simple construction, with outside-packed plungers. The following table gives sizes used for this type of pump.

THE WORTHINGTON PACKED-PLUNGER PUMP

Scranton Pattern—For 250 Pounds Water Pressure.

Diameter of Steam Cylinders.	Diameter of Water Plungers.	Length of Stroke.	Gallons per Revolution.	Revolutions per Minute.	Gallons per Minute.	Size of Pipes for Short Lengths to be Increased as Length Increases.				Approximate Space Occupied, Feet and Inches.			
						Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Length.		Width.	
14	8½	10	9.56	54	516	2½	3	8	6	9	8	3	2
16	8½	10	9.56	54	516	2½	3	8	6	9	9	3	10
18½	8½	10	9.56	54	516	3	3½	8	6	9	10	4	0
16	10¼	10	13.95	54	753	2½	3	10	8	10	9	3	10
18½	10¼	10	13.95	54	753	3	3½	10	8	10	9	4	0
18½	12	10	19.16	54	1035	4	3½	12	10	11	1	4	0
20	12	10	19.16	54	1035	2½	5	12	10	11	2	4	2
17	8½	15	14.14	40	565	4	3½	8	6	10	5	3	11
20	8½	15	14.14	40	565	2½	5	8	6	10	6	4	2
17	10¼	15	20.83	40	833	4	3½	10	8	11	6	3	11
20	10¼	15	20.83	40	833	4	5	10	8	11	8	4	1½
20	12	15	28.78	40	1151	5	5	12	10	11	9	4	3

A *sinking pump* is one used in mines for dropping into a shaft which has to be freed of water collected during a time of disuse or which has come in when a subterranean stream has been cut into. The pump is subject to rough usage and must be applied quickly and easily to the timbers of the shaft. Fig. 142 shows one form of sinker as built by the Prescott Company. This pump is mounted on a frame which can be hung from the mine timbers. The working parts are protected from injury from falling objects by the projections of the cylinder and steam chest. The chains or rods connecting the link serve to support the pump from the derrick. The water-valve boxes are accessible for repairs and the outside-packed plungers make it possible to be sure that there is no leakage from side to side. When electricity is applicable the electric-

sinking pump of the form shown in Fig. 143 is used. This pump is driven by the motor through a double reduction gear and the plungers are of the form mentioned earlier. The method of lowering, the protection of the parts from mechanical injury and the motor from water, as well as the arrangement of the valves for quick inspection are clearly seen. In each of these sinking pumps the suction is connected to the bottom of the pump and the discharge may be taken from the right

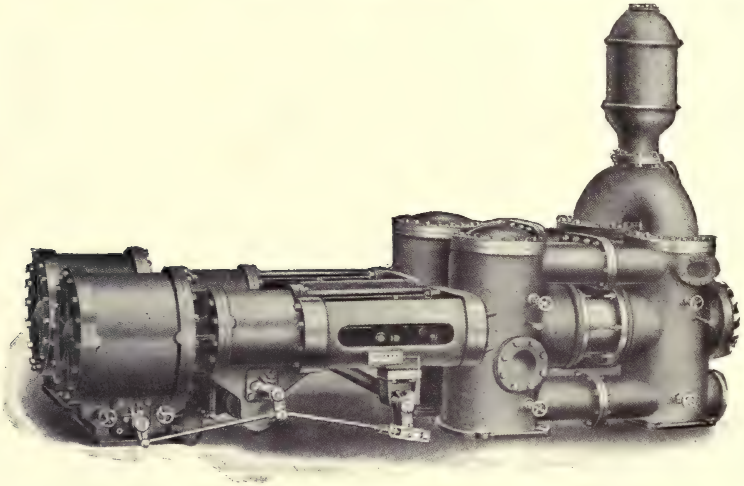


FIG. 141.—Mine Pump Scranton Pattern.
(Size 16 and 25×14×15.)

or the left. These pumps are so built that they may run if flooded by water, as such a condition may arise at any time. The table on page 151 gives the sizes used with the Knowles electric duplex sinking pump.

THE KNOWLES VERTICAL DUPLEX ELECTRIC SINKING PUMPS

Double-Acting Outside Center-Packed Plunger Pattern.

This pump is light, compact, efficient, and of good capacity, and not liable to damage from moisture or hard usage. The entire motor mechanism is inclosed in a tight casing, but every

Diameter of Plunger, Inches.	Stroke, Inches.	Vertical Head in Feet.	Revolutions per Minute.	Gallons per Revolution.	Gallons per Minute.	Suction Pipe, Inches.	Discharge Pipe, Inches.	Ratio of Pump Gears.	Ratio of Motor Gears.	H.P. Motor Required.	Dimensions over all in Feet and Inches.		
											Width.	Depth.	Height.
3	6	400	55	.73	40	3	2	3.5 to 1	To suit speed of motor	10	3 0	2 9	8 8
4	6	200	80	1.30	104	4	3	2.7 to 1		7½	3 0	2 9	8 8
4	6	500	80	1.30	104	4	3	3.7 to 1		20	3 8	3 6	9 4
5	6	300	75	2.04	153	5	4	3.7 to 1		20	3 8	4 2	10 2
5	6	400	80	2.04	163	5	4	3.7 to 1		30	3 8	4 2	10 4
6	6	200	86	2.93	252	6	5	3.7 to 1		20	3 9	4 2	10 4
6	6	250	86	2.93	252	6	5	3.7 to 1		25	3 9	4 2	10 4
6	8	400	62	3.91	242	6	5	3.5 to 1		50	4 6	4 10	11 2
6½	8	300	70	4.59	321	6	5	3.5 to 1		50	4 6	4 10	11 2
7	8	300	68	5.32	362	8	6	3.5 to 1		50	4 6	4 10	11 2
8	8	300	58	6.96	403	8	6	3.5 to 1		50	4 6	4 10	11 2

part of motor and pump is readily accessible for examination or repairs. This apparatus will stand hard usage without injury. The main frame is designed especially to receive the specified type and size of motor. As the plungers and piston rods are packed from the outside, any leakage is readily noticeable. Each pump is furnished with a discharge air chamber. In sending inquiries or orders, always state as fully as possible the intended service and requirements.

Centrifugal pumps are now being used for this purpose at times.

The Water-Works Pump will be considered in a later chapter in more detail, but at this point two simple forms will be described. It is this kind of pump which has reached the highest form of development, because such machines are used for the sole purpose of raising water in large quantities and continuously. They may be of the direct-acting type (Fig. 144) or the fly-wheel type (Fig. 145). They are usually compound or triple expansion on the steam end, and precautions are taken to cut down all losses. The pumps are made with great refinement of parts, as they are usually handled by skilled engineers.

In most cases there is a possibility of adjusting the steam-

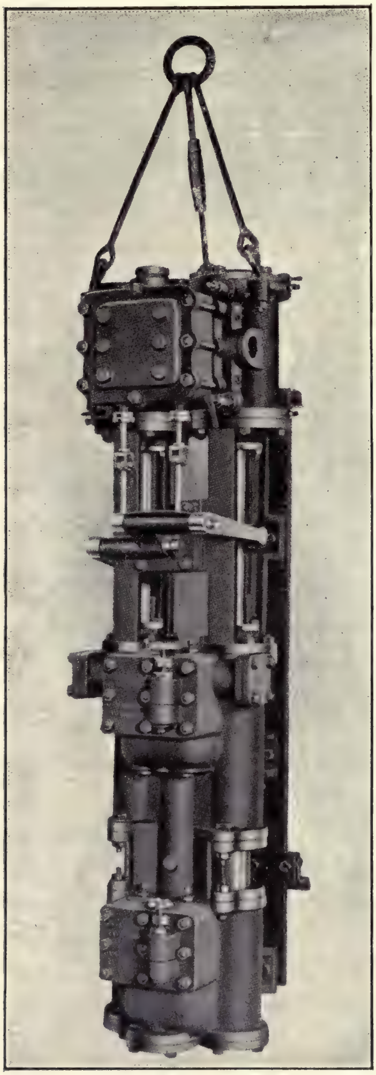


FIG. 142.—Prescott Steam Sinking Pump.

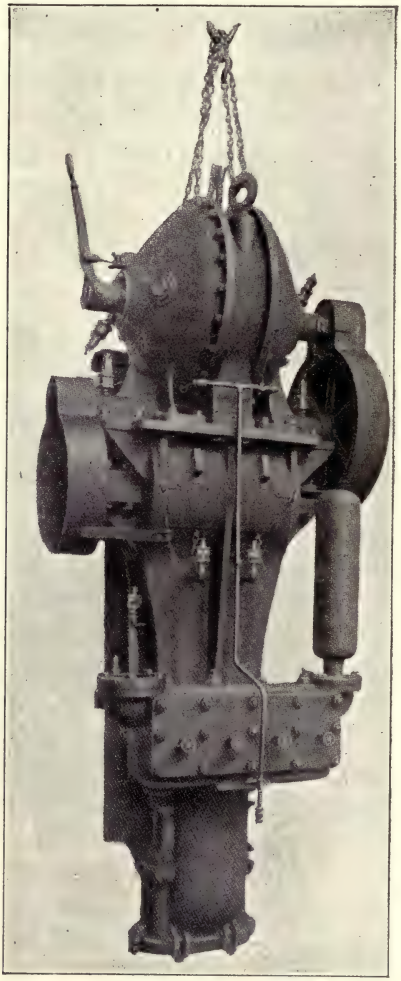


FIG. 143.—Electric Sinking Pump. (Size 6×6.)

valve gearing, and the water valves may be examined with ease. These pumps vary in size from those handling 50,000

gallons per twenty-four hours to those handling 30,000,000

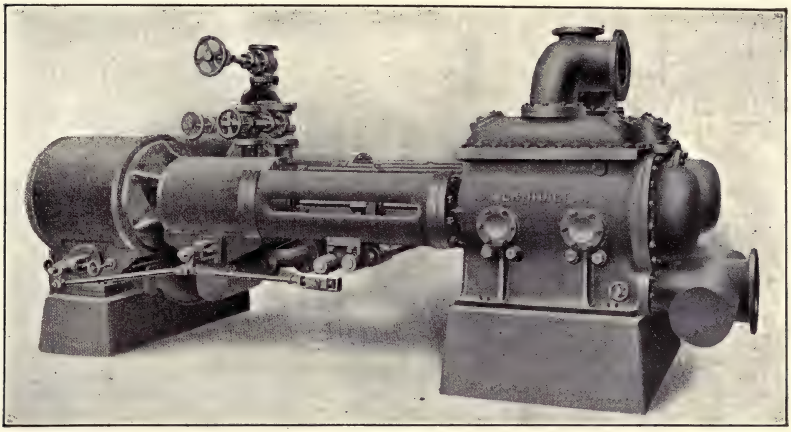


FIG. 144.—Direct-Acting Water-Works Pump.
(Size 16 and $25 \times 14\frac{1}{2} \times 18$.)

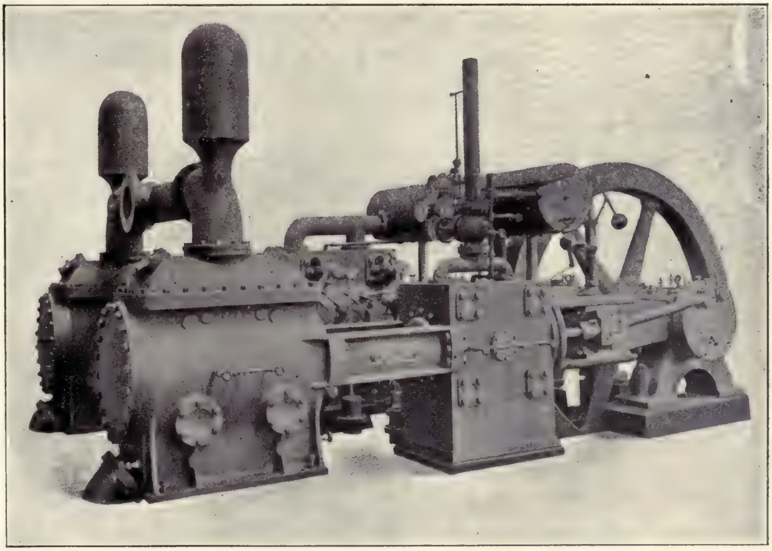


FIG. 145.—Water-Works Pump.

gallons in the same time. Such pumps are usually rated in this manner—gallons per twenty-four hours.

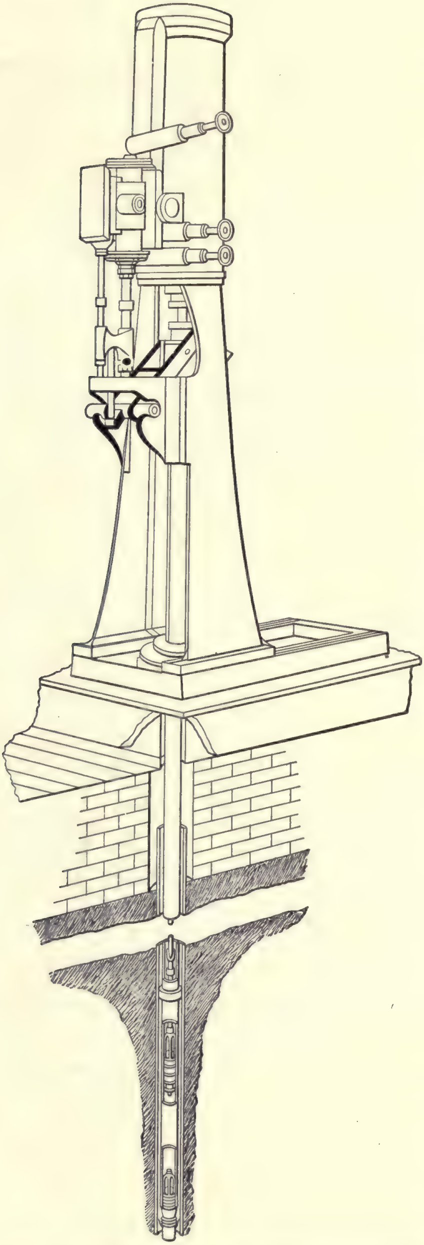


FIG. 146.—Fairbanks-Morse Deep-Well Pump.

In connection with water works it becomes necessary to lift water from artesian or deep wells, and for this work special *deep-well pumps* have been designed. These consist usually of a steam cylinder mounted at the top of the deep well (Fig. 146). The piston rod of such a pump is usually made into a plunger passing through a stuffing box at the base of the pump frame. This plunger is then connected to a bucket piston by wooden rods with iron joints. The bucket may be several hundred feet below the surface. This distance is fixed by the height at which the water stands in the well when the pump is in operation. The foot valve is placed at the end of the pipe line which forms the pump barrel. This foot valve is placed in position by the pump rods, although in some cases it is lowered into position by other means, its weight holding it after placing. The pump bucket as shown by the figure is packed by means of cup

leathers and the foot valve and head valve shown are of the ball type. This last section of pipe casing seen in the figure may be of heavy drawn brass tubing to make a proper pump barrel. In arranging the plunger at the top of the wooden sucker rods, it should be made of net area equal to one-half the net area of the bucket so as to discharge water on both strokes of the pump.

The pump frame is so constructed that it may be moved over from the top of the well in order that the rods or piping can be taken up. This is done by hoisting the rods or casings until one section is above ground, then clamping the line by the bottom section until the upper section is unscrewed and removed, when the operation is repeated. For this reason it is necessary to construct the pump house over a deep well with sufficient head room to permit the removal of casings or rods; this is usually about 20 or 25 feet. The tables below give data from catalogue of the Fairbanks-Morse Co. in regard to their pumping engines.

THE FAIRBANKS-MORSE ARTESIAN-WELL ENGINE

This engine is placed directly over the well, and the piston rod is continued to the required depth and connected to the pump piston. The steam valve is perfectly controlled, and the speed of the engine on both up and down strokes is uniform. The apparatus may be run at a high speed without excessive shock or jar.

These engines will pump from the deepest wells, forcing the water in a steady stream into an elevated tank or other reservoir. To remove the pump rods and pistons, the bolts which connect base to the frame are loosened and the steam cylinder and uprights are drawn back on the base by a screw. The upper displacing cylinder discharges one-half the volume pumped on the down stroke, thus tending to balance the machine and insure a smooth and easy action.

Size of Engine.		Size of Pipe.		Floor Space in Inches.
Diameter of Steam Cylinder.	Length of Stroke.	Steam.	Exhaust.	
6	18	1	1½	48 × 20
8	24	1¼	1½	51½ × 23
10	36	1½	2	62 × 25
12	36	1½	2	62 × 25
14	36	2	3	73 × 34½
16	36	2	3	73 × 34½

BRASS ARTESIAN-WELL CYLINDER

Inside Diameter.	Length of Stroke.	Capacity per Stroke in Gallons.	Outside Diameter of Caps.	Top and Bottom Connecting Pipe in Inches.
2¼	18	.31	3 9/16	2½
2¾	18	.46	3⅞	3
3¼	18	.64	4 7/16	3½
3¾	18	.86	5⅛	4
2¾	24	.61	3⅞	3
3¼	24	.86	4 7/16	3½
3¾	24	1.84	5⅛	4
4¼	24	1.47	5¾	4½
4¾	24	1.84	6¼	5
5¾	24	2.68	7¼	6
3¼	36	1.29	4 7/16	3½
3¾	36	1.71	5⅛	4
4¼	36	2.21	5¾	4½
4¾	36	2.76	6¼	5
5¾	36	4.02	7¼	6

TABLE OF WOOD SUCKER RODS

Diameter of Rod in Inches.	Adapted for Working Barrels of a Diameter
1⅝	2¼ to 4¼
2¼	4¼ to 5¾
3½	5¾ to 7¾

A gas engine geared to a deep-well pump is shown in Fig. 147, in which the same points are to be noted as in Fig. 146, while Fig. 148 illustrates the method of using a horizontal cylinder for this purpose, as proposed by Davidson.

A form of deep-well power pump used in the West is shown by Fig. 149. In this, the *Luitwieler pump*, motion is given

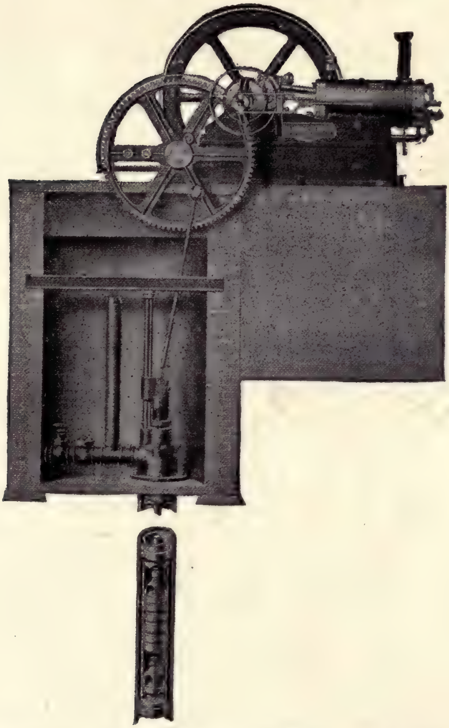


FIG. 147.—Gas Engine Drive for Deep-Well Pump.

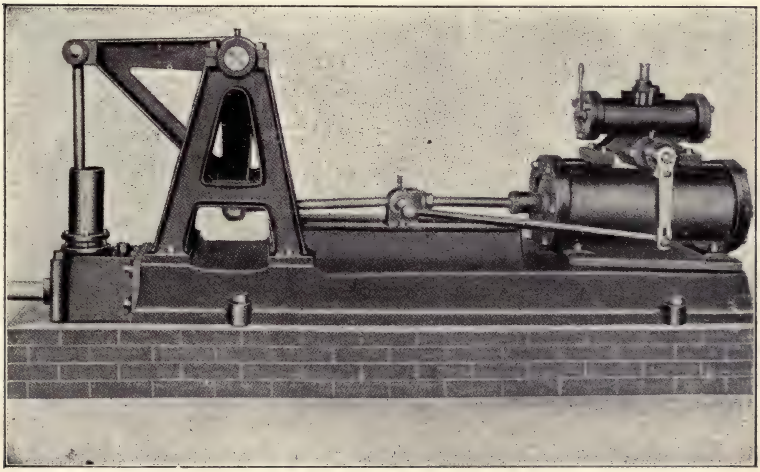


FIG. 148.—Davidson Horizontal Steam Cylinder for Deep-Well Pump.

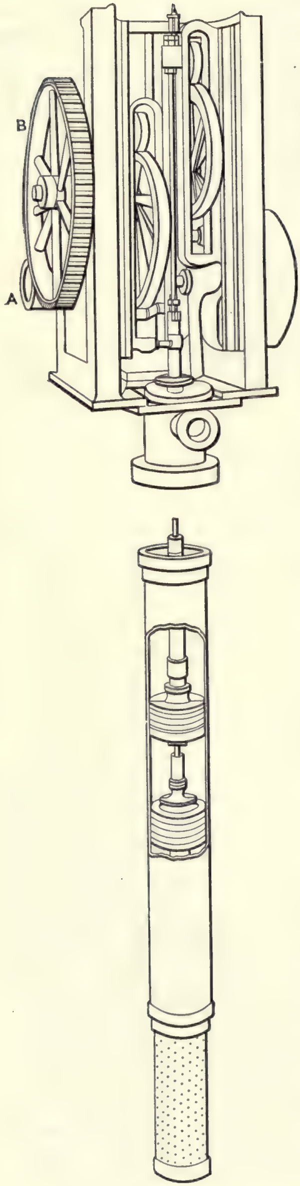


FIG. 149.—Luitwieler Deep-Well Pump.

to the spur wheel A by a belt, steam engine, gas engine, or electric motor, and this drives the gear B. On the gear shaft are two cams arranged opposite each other and so made that they have a lifting motion during 53 per cent of the rotation, while the descending motion occupies 47 per cent of a revolution. These cams drive cross-heads by means of frictionless rollers. The left-hand cross-head is attached at its lower end to a hollow pump rod and the right-hand one at its upper end to a solid rod passing through the hollow one. The cross-heads are guided to prevent turning. The hollow rod passes through a stuffing box and has at its upper end a stuffing box for the solid rod. These rods are each attached to a bucket piston as shown by the figure. The outer rod is made of larger bore than the diameter of the solid rod so that there is no friction, provided the rod does not whip and is vertical. When the lower piston is descending the upper one is ascending and forcing the water out, while water will flow in between the two pistons to fill the space formed there. Before the upper piston reaches the top of its stroke the lower piston starts to move upward and when the upper piston begins to descend the lower piston has sufficient speed upward to keep the column of water in the pipe in motion at the same speed.

The water lifted by the lower bucket is forced through the upper bucket. Before the lower bucket ceases to move upward the upper one has completed its down stroke and has started

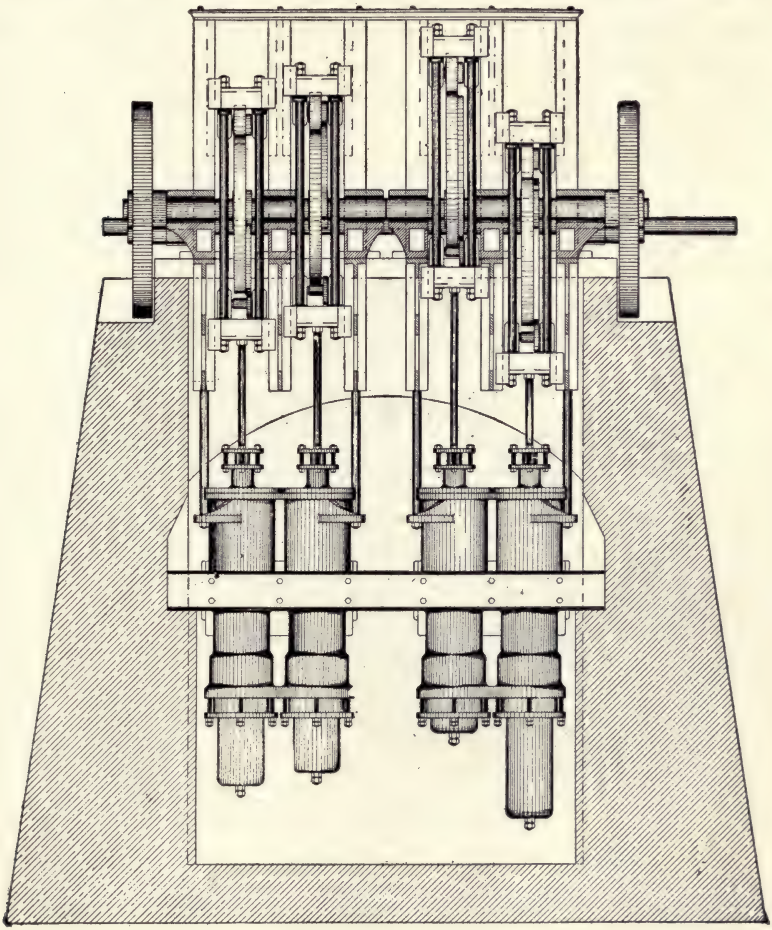


FIG. 150.—Luitwieler Quadruplex Pump.

back again ready to continue the upward motion of the water in the pipe as soon as the lower bucket begins to diminish its upward rate and reverse its action.

The buckets, rods, and cross-heads balance each other quite well, so that there is a fairly uniform torque on the driving

mechanism. The action is different from the ordinary deep-well pump and its differential plunger in that not only the stream above ground is moving but also that in the well barrel. The pump lifts water on the suction side during one-half revolution, as in the case of the ordinary pump, but the discharge of this through the well barrel is distributed over the whole revolution. If it were possible to put the upper plunger of the deep-

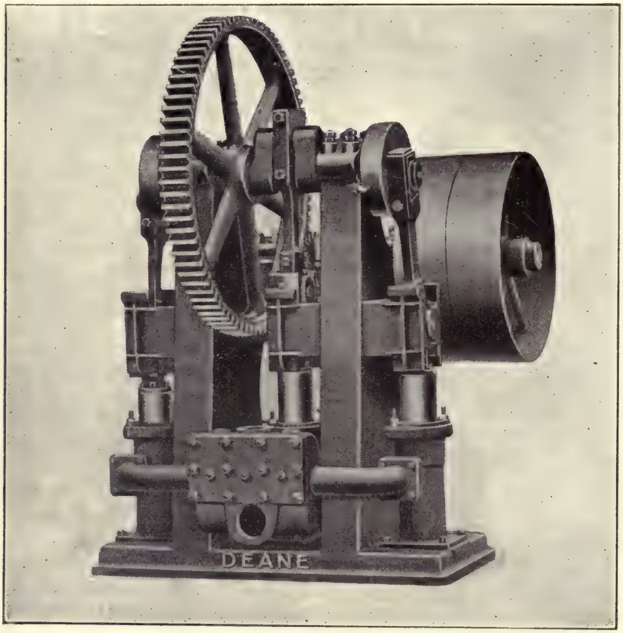


FIG. 151.—Triplex Pump.
(Size 5×8.)

well pump down near the working barrel the same action would occur in that pump. The arrangement of cams may be such that the motion of the water is uniform, the accelerating period of the motion of one occurring during the time in which the other is changing its motion.

The same arrangement of cams may be applied to duplex or quadruplex pumps, where the motion given by the ordinary crank is not sufficiently uniform. Such an arrangement is

shown in Fig. 150. In this case the effort is made to submerge the pump barrels to cut down all suction lift.

This leads to a large class of pumps known as *power pumps.* Power pumps are those in which the pump proper is driven through gears or belts. These pumps are of various forms, some horizontal, some vertical, some with plungers, some with

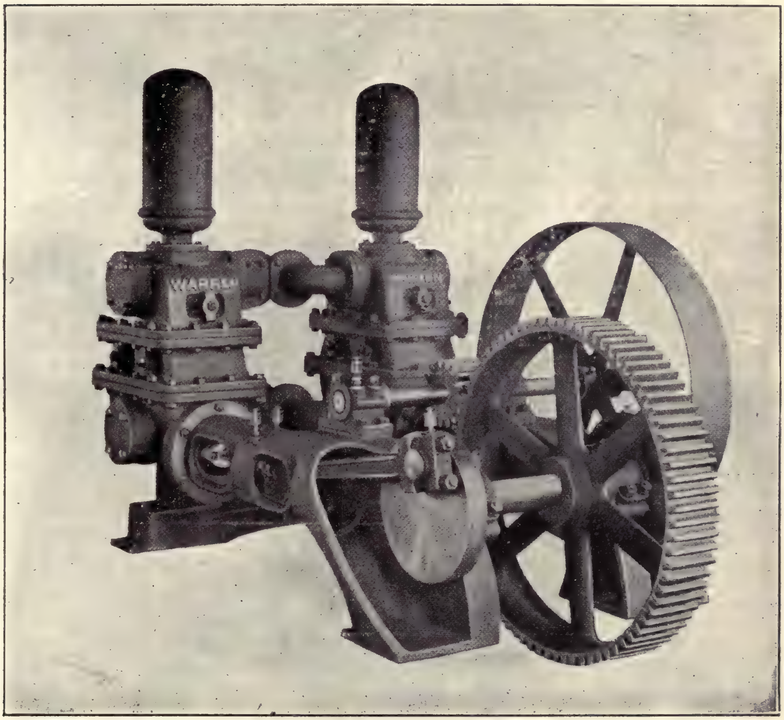


FIG. 152.—Horizontal Duplex Power Pump.
(Size 6×12.)

pistons, some with single cylinders, some with double cylinders, and some triplex. The name triplex usually designates a vertical, three-cylinder pump. In many cases it is of the plunger type. Fig. 151 illustrates this form. The large valve case with a simple arrangement for examining the valves, the inlet and outlet openings, the short connections to the cylinder, the method of supporting the side thrust, the method of

packing, and the arrangement of cranks and gears are clearly shown. The sizes of such pumps are given in the table below.

DEANE TRIPLEX VERTICAL SINGLE-ACTING POWER PUMP

Size.		Maximum Water Pressure.	Capacity.			Pipe Sizes.		Tight and Loose Pulleys.			Ratio of Gearing.	Approximate Dimen. in Feet and Inches.		
Diam. of Plungers.	Length of Stroke.		Gallons per Revolution	Revolutions per Minute.	Gallons per Minute.	Suction.	Discharge.	Diameter.	Face.	Belt.[1]		Length.	Width.	Height.
3	8	290	.73	45	33	3	2½	30	5½	D	4.7–1	3 10	3 9	5 7
4	8	290	1.30	45	58	3	2½	30	5½	D	5.0–1	4 8	4 11	6 5
4	8	270	1.30	45	58	3	2½	30	5½	D	4.7–1	3 10	3 9	5 7
4½	8	290	1.65	45	74	4	3	30	7½	D	5.0–1	4 8	4 11	6 5
5	8	254	2.04	45	91	4	3	30	7½	D	5.0–1	4 8	4 11	6 5
5	8	180	2.04	45	91	4	3	30	6½	D	4.7–1	3 10	3 9	5 7
5½	8	209	2.46	45	110	4	3	30	7½	D	5.0–1	4 8	4 11	6 5
5½	8	150	2.46	45	110	4	3	30	6½	D	4.7–1	3 10	3 9	5 7
6	8	200	2.93	45	132	4	3	30	7½	D	5.0–1	4 8	4 11	6 5
6	8	125	2.93	45	132	4	3	30	6½	D	4.7–1	3 10	3 9	5 7
6½	8	150	3.44	45	154	[2] 5	[2] 4	36	7½	D	5.0–1	4 8	4 11	6 5
6½	8	107	3.44	45	154	[2] 5	[2] 4	30	6½	D	4.7–1	3 10	4 1	5 7
7	8	150	4.00	45	180	[2] 5	[2] 4	36	8½	D	5.0–1	4 8	4 11	6 5
7	8	92	4.00	45	180	[2] 5	[2] 4	30	7½	D	4.7–1	3 10	4 1	5 7
8	8	100	5.22	45	235	[2] 6	[2] 5	36	8½	D	5.0–1	4 8	4 11	6 5

[1] D indicates double belt. [2] Flanged connections.
Above pumps may be safely run at somewhat greater speed than listed.

The pumps listed above have outside-packed plungers which work through deep-stud gland stuffing boxes. All sizes have cross-heads for guiding the travel of the plungers, and approved means provided for taking up wear. The bearings are of ample diameter and length. The gearing is a special mixture of steel and iron, and the teeth are machine cut. The connecting rods are provided with adjustable boxes at both ends, the adjustment being made by means of a wedge and screw.

Pumps of this type are used for general service and tank service; as boiler feeders, elevator pumps, and water works.

This type of pump is built with 3-inch, 6-inch, 8-inch, and

12-inch stroke, each class having the same general characteristics as described above.

Fig. 152 shows a horizontal duplex power pump with pistons. There are many different forms of this type of pump, but these two will illustrate the ordinary styles.

Fig. 153 illustrates the power head for a deep-well pump,

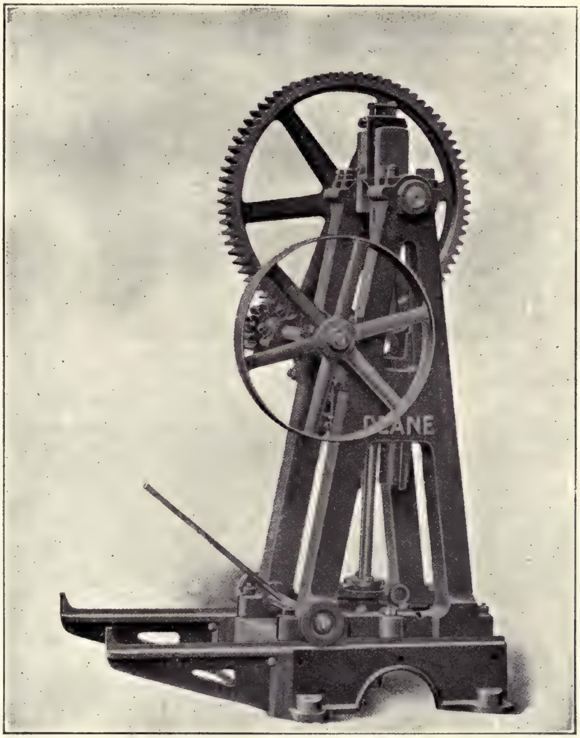


FIG. 153.—Power Head f2r Deep-Well Pump.

giving the method of operating such a pump with belting. An electric motor could be attached by a coupling or belt to the shaft carrying the belt pulley. The figure shows the method of removing the top gearing from the well when it is necessary to remove rods or casing.

A new form of power pump usually driven direct from an electric motor without the use of gears to cut down the

speed is the *express pump*. Such a pump, Fig. 154, is arranged to run at high rotative speeds of about 200 to 300 R.P.M. The *Riedler pump* of the Allis-Chalmers Company is arranged to permit this high speed by having a clear direct passage for the water through large valves and by making the discharge valve positive in its closing. The water enters through the annular space AB from the vacuum air chamber C. It is then discharged into the air chamber K through the valve E. This

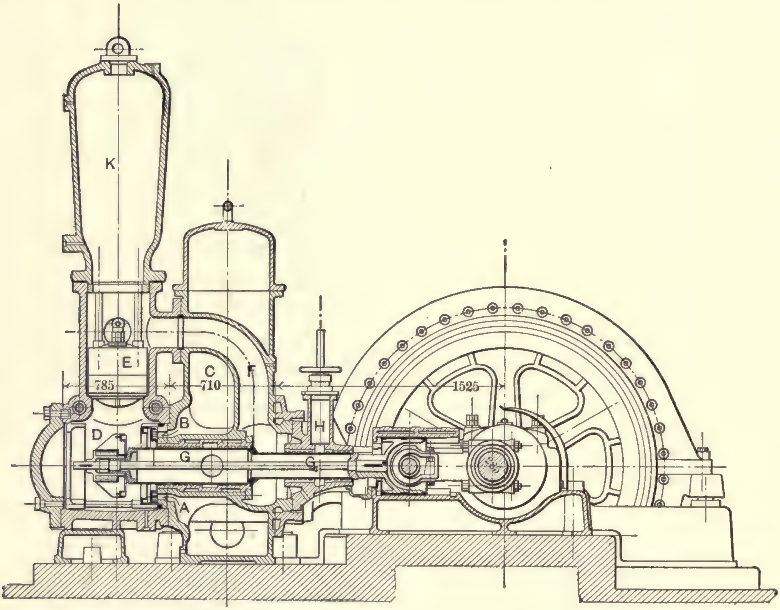


FIG. 154.—Allis-Chalmers' Duplex Riedler Express Pump

valve is closed positively by means of an eccentric on the shaft. The plunger G_1 is of one-half the area of G so that water is passed back and forth through F, giving a discharge on each stroke, although there is only one suction stroke to each revolution.

The ring D, carried from the back of the plunger, closes the annular valve when the plunger reaches the end of the suction stroke, thus cutting down the chance of slip from the slow action of the valve.

The suction air chamber C insures a supply of water at all times.

The express pumps of Riedler were among the earliest of this type, but in later years there have been many new forms introduced. In some cases the valves have been spring controlled, eliminating the necessity of the valve gearing.

The simplex pump, the water-works pump, the centrifugal and air-lift pumps, air pumps for condensers, and other special pumps will be considered in later chapters.

CHAPTER IV

SIMPLEX PUMPS

THE pump containing one steam cylinder and one water cylinder is known as a simplex pump to distinguish it from the duplex pump, so largely described in the previous chapter. This type was the original form used by Worthington, and it has been worked on by many inventors. It is the purpose of this chapter to describe a number of the well-known types which will show the general manner of solving the problem of getting a definite reversal of the pump at the end of the stroke no matter what the speed of the pump is. This has been the aim of all simplex-pump designers. It is a simple matter to reverse the action when the pump is moving rapidly, but for slow action there must be some auxiliary apparatus, as the spring of the first Worthington pump, Fig. 55.

The simplex pump is undoubtedly more complex than the duplex pump, but with it there is little danger of short stroking. In the duplex pump the valves must be carefully adjusted, as the motion of one piston controls the action of the valve of the other side. The simplex pump in many cases may be run at higher speeds than are common with duplex pumps. The loss due to the clearance space is a total loss in direct-acting pumps, as there is no expansion, and for this reason the clearance should be made as small as possible and no short stroking should be allowed.

The Cameron Pump (Fig. 155) is one of the simpler forms of these pumps. As shown in the figure the *B*-valve *G* is admitting steam to the right-hand side of cylinder *A* and the piston is being driven to the left. When the piston *C* overruns the steam passage at its left end the exhaust steam is cushioned partially as it can only pass through a groove at the

top of the cylinder to the passage. The piston C strikes the pin on the cylindrical valve I, allowing the steam to the left of the piston F to escape through E into a passage leading to tht exhaust. The steam which has leaked into the space ae the right of the auxiliary piston F, through the small hole in the center of the head, drives this to the left, moving with it the valve G, and reverses the pump. When the piston C moves to the right, live steam entering behind I through the passage K

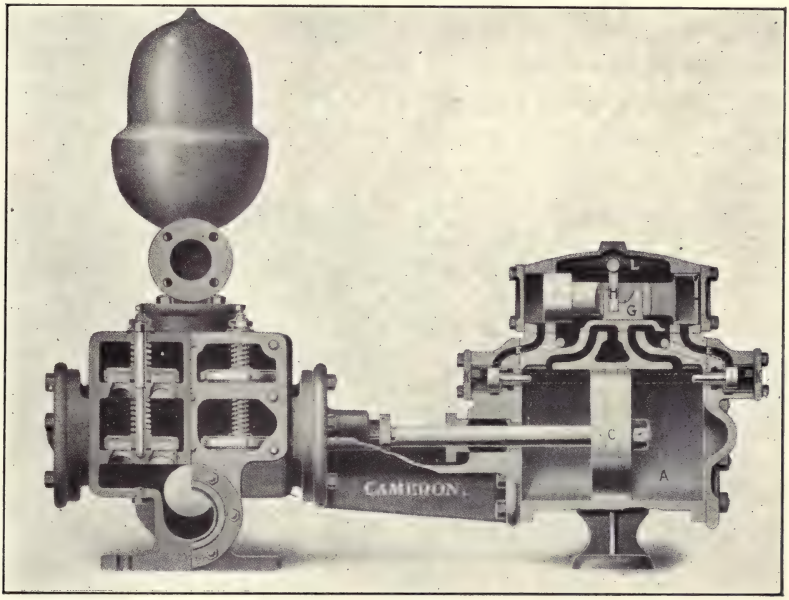


FIG. 155.—Cameron Pump.

forces the valve I over, closing the passage to the left of the cylinder in which F travels while the steam which leaks through the small hole in the end of F builds up the pressure for the next reversal, when the right-hand valve I is opened. The auxiliary piston F controls the valve G. It is made of two hollow pistons with holes in the ends. The steam which leaks through these openings represents the cost of operating the valve. This may be an appreciable amount, as the steam may fill the space at the left-hand end in the figure before

the exhaust occurs. With a small hole, however, there need not be much steam used, as it is only the steam on the smaller end which is needed to reverse the valve. The handle *H* projecting from the side of the steam chest serves to move the valve back and forth in starting.

The figure shows the arrangement of the suction and discharge valves. These valves are controlled by springs, and by taking out a common spindle the valves on one side may be removed. The clear passage for the suction is seen and the short length of this pump in comparison with its stroke is important where space is necessary.

The table on p. 169 gives the sizes of this pump used for boiler feeding as arranged by the makers.

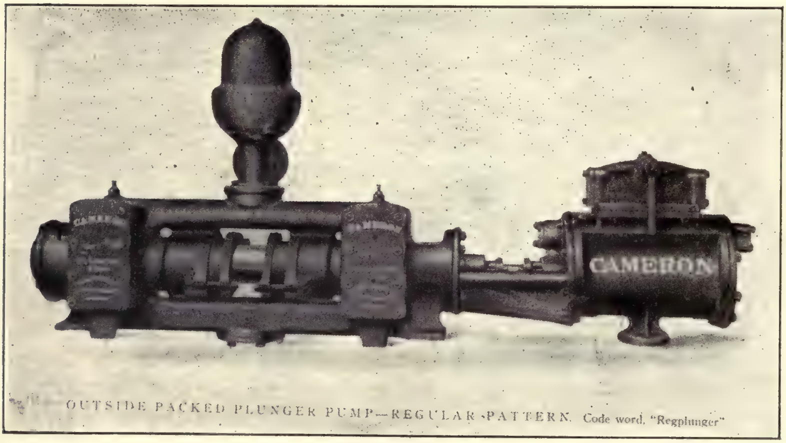


FIG. 156—Cameron Pump.

CAMERON PISTON PUMP—REGULAR BOILER-FEED PATTERN

The main difficulty met with in fixing on the proper size of pump to recommend is that the horse power of the boiler for which the pump is required is about all the information furnished. The expression "horse power," as applied to boilers, is a very indefinite term; what should be given, if

Size Number.	Price, with Iron Water Cylinder and Steel Piston Rod.	Price, with Water Cylinder Lining, Piston and Piston Rod of Composition.	Diameter of Steam Cylinder, Inches.	Diameter of Water Cylinder, Inches.	Stroke Piston, Inches.	Capacity at Ordinary Speed per Minute, Gallons.	Boilers, in H.P. they will supply at ⅓ Ordinary Speed based on 30 lbs. of Water per H.P. per Hr.	Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Discharge Pipe.	Floor Space, Inches.	W'ght.
0	$80	$85	3½	2	4	8	40	⅜	½	1¼	1	32×9	136
1	120	125	4	2	6	12	60	⅜	½	1¼	1	40×10	210
2	140	150	5	2½	6	18	90	½	¾	1½	1¼	40×11	254
2*a*	150	165	5	3	7	28	140	½	¾	2	1½	47×12	347
2*b*	160	170	5	3½	7	38	190	½	¾	2½	2	47×14	355
3*b*	210	225	6	4	7	50	250	¾	1	2½	2	47×14	422
3*c*	210	225	6	3½	12	50	250	¾	1	3	2½	58×17	520
3*d*	230	245	6	4	12	65	325	¾	1	3	2½	58×17	525
5*a*	280	310	7	4½	12	80	400	¾	1	3	2½	58×18	725
5*b*	325	350	7	5	13	100	500	1	1½	4	3	63×20	1117
......	375	400	8	6	13	150	750	1	1½	4	3½	64×21	1202
......	445	490	10	7	13	200	1000	1¼	2	5	4	66×24	1770
......	500	530	12	8	13	261	1300	1½	2½	5	5	73×26	2010
......	....	....	14	9	18	330	1600	2	3	6	5	81×30	3125
......	....	....	14	10	18	400	2000	2	3	6	5	81×30	3300
......	....	....	16	10½	18	450	2300	2½	4	8	6	90×37	4827
......	....	....	16	12	20	587	3000	2½	4	10	8	103×41	5140

possible, is the quantity of feed water required, and a pump which will supply this quantity at about one-half its rated capacity at ordinary speed will be right for cold water, and say one-third speed for hot water.

In feeding hot water the pump should be placed below the source of supply. The first six sizes are furnished with hand-lever attachment when so desired.

An outside-packed type of the Cameron pump is shown

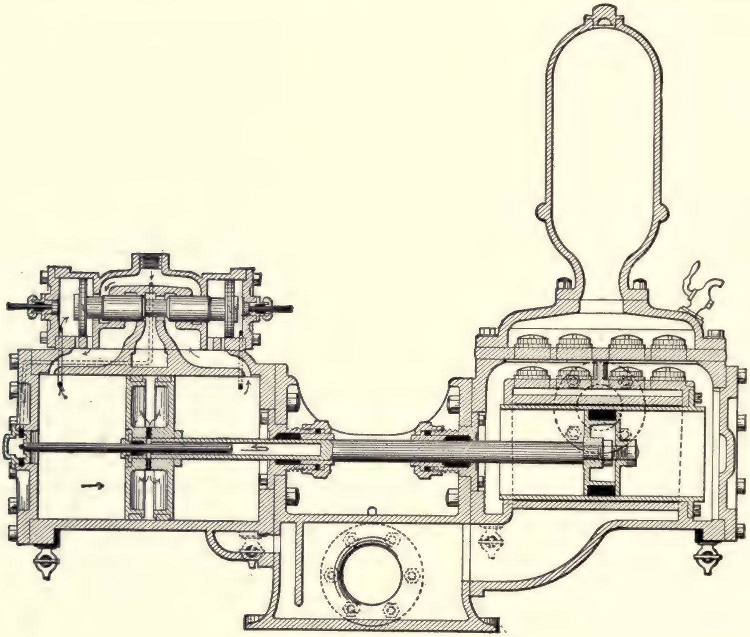


FIG. 157.—Marsh Steam Pump.
(8×5×10, 8×5×12, 10×6×12.)

in Fig. 156. The view not only shows the type of water end, but also shows the reversing valve cylinders on the steam cylinder heads and the reversing handle.

The Marsh Pump (Fig. 157) is another one using a steam-thrown valve. In the picture shown the piston travels to the right while the steam enters the cylinder through the annular space around the spindle valve and passes into the steam

passage. Should the water resistance on the water piston decrease and the steam piston respond suddenly to this, the pressure in the cylinder would fall and with it the pressure on the left of the disc on the left end of the spindle-slide valve. The pressure on the right-hand side of this disc would then move the spindle, cutting down the steam supply and throttling the exhaust, thus dampening the action of the steam piston.

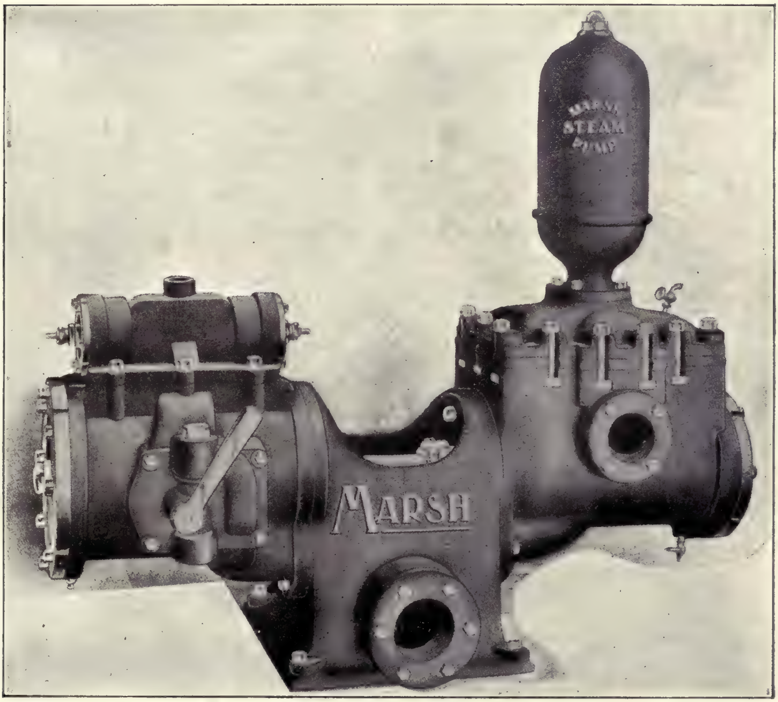


FIG. 158.—Marsh Pump.

When the piston reaches the end of its stroke it overrides the small port leading to the right-hand end of the valve piston, and the live steam which is introduced into the central space between the two parts of the steam piston, then passes behind the right-hand side of the right disc on the valve spindle, and forces it to the left. This admits steam to the right-hand end of the steam cylinder and opens the left-hand end to the

exhaust. It is clear from the figure that the two faces of the right-hand disc are exposed to exhaust pressure and the left-hand disc has live steam on each side; the valve is thus balanced until the central part of the main piston covers the auxiliary passage, admitting live steam on one side and thus destroying the balance in such a manner that the valve shifts readily. The dotted lines show the passages carrying steam to the central part of the piston and to the valve case.

The balanced feature of the valve means that the valve is adjusted for speed so that the pump will not race.

This company sometimes builds the steam piston with the central space as wide as the stroke so that steam may be introduced into this space by a connecting passage in the cylinder casting, saving the packed rod. This method means a much longer steam cylinder and also considerable loss from steam condensation.

The table below gives the sizes of these pumps:

MARSH STANDARD BOILER FEED-PUMPS

FOR HIGH PRESSURE. REGULAR LINE

Steam Cylinder.	Water Cylinder.	Stroke.	Gallons per Stroke	Gallons per Minute.	Gallons per Hour.	Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Floor Space.	Horse-power.	Weight, lbs.
7	4½	8	.55	66	4,000	1	1¼	3½	2½	15×40	400	650
7½	4½	10	.68	75	4,500	1	1¼	3½	2½	18×48	450	900
8	5	10	.85	85	5,100	1	1½	3½	2½	18×48	500	1000
8	5	12	1.02	100	6,000	1	1½	4	3	18×52	600	1150
10	6	12	1.46	146	8,750	1¼	1½	4	3	18×52	1000	1300
12	7¼	12	2.14	216	13,000	1½	2	5	4	19×56	1500	1500
12	8	12	2.61	260	15,600	1½	2	5	4	19×56	1650	1500
14	8½	12	2.94	295	17,700	2	2½	6	5	22×62	2000	2400
16	10	16	5.44	408	25,000	3	3½	8	7	24×76	2500	4800
16	9¼	16	4.65	350	21,000	3	3½	8	7	24×76	2200	4800
16	10	20	6.80	450	27,000	3	3½	8	7	24×76	2700	5000
16	9¼	20	5.81	400	24,000	3	3½	8	7	24×76	2400	5000
20	12	20	9.78	588	35,000	3½	4	8	7	26×93	3500	6500

Capacity. Capacity rating above is intended to represent maximum service recommended. It is always advisable to run pumps at moderate speed, consequently select sizes large enough to meet ordinary requirements easily. Reserve power in a pump is quite as important as in a boiler or engine.

The small tappets at each end of the valve casing are used to start the pump in case it does not operate. Fig. 158 gives an outside view of this pump while the method of attaching the suction is seen in the two figures. Steam may be exhausted into the base where the suction enters by turning the handle shown in Fig. 158. The exhaust steam heats the feed water and is thus saved for useful application.

One of the oldest of the simplex pumps is the Knowles pump. The Warren steam pump is quite similar to it, and its action will be described. The operation of this pump is

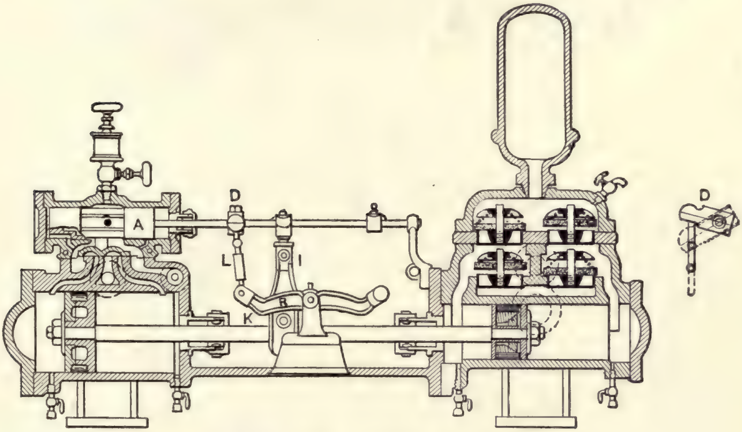


FIG. 159.—Knowles Pump.

shown in Fig. 159. When the steam piston reaches the left end of its stroke the roller *K* strikes the lever *R*, forcing up the rod *L*, which is attached to an arm *D*, projecting from the rod of the auxiliary piston *A*. This rotates *A* through a small angle and brings a groove in *A* extending to the right end over a small steam passage while a groove on the left end of *A* is brought over an exhaust passage. This forces the auxiliary piston to the left, moving the *B*-valve into the position shown, reversing the pump. At the right-hand end of the stroke the roller *K* strikes the other end of *R*, pulling the rod *L* downward. This rotation brings the groove on the right end of *A* over an exhaust passage, while that on the left comes opposite a steam

passage and the auxiliary piston is driven to the right. Should the rotation fail to reverse the pump, the rod *I* would hit the

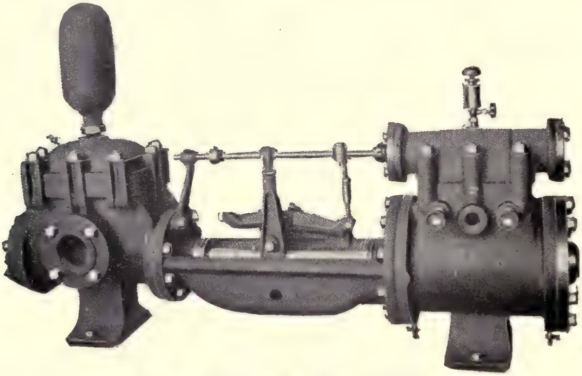


FIG. 160.—Knowles Boiler Feed Pump.
(Size 10×6×12.)

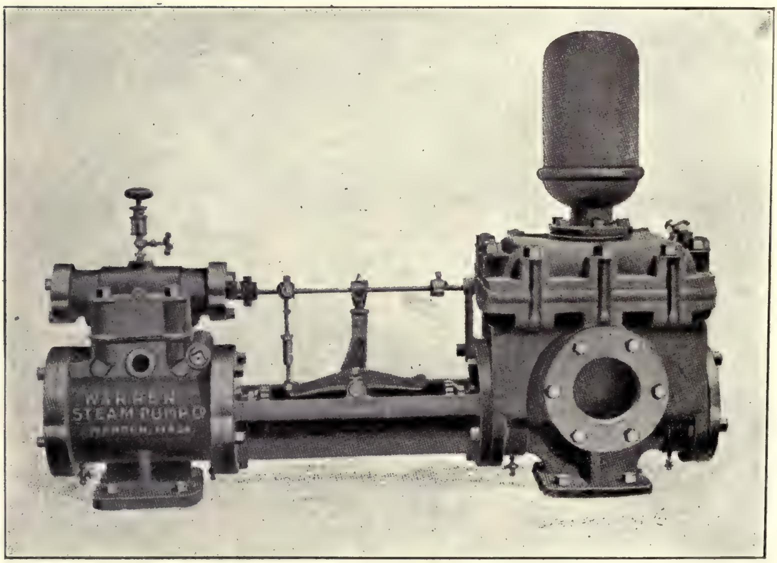


FIG. 161.—Warren Pump.
(Size $7\frac{1}{2}\times7\frac{1}{2}\times10$.)

tappets on the rod from the auxiliary piston and drive the valve to its opposite position.

The roller may be raised and lowered. This with the adjustment in the connecting rod L makes it possible to adjust the point of reversal.

The Knowles heavy pattern boiler-feed pump is shown in Fig. 160, while Fig. 161 shows a Warren light-service pump.

The table below gives sizes of the Knowles pump.

THE KNOWLES HORIZONTAL BOILER-FEED OR PRESSURE PUMPS

Piston Pattern—For Hot or Cold Water or Other Liquids

All parts of this pump are interchangeable, and can therefore be readily duplicated in case of accidental breakage or unusual wear.

REGULAR PATTERN. FOR 125 POUNDS WATER PRESSURE.

Number.	Steam Cylinder, Inches.	Water Cylinder, Inches.	Stroke, Inches.	Gallons per Stroke.	Capacity per Minute at Maximum Speed.		Steam Pipe, Inches.	Exhaust Pipe, Inches.	Suction Pipe Inches.	Delivery Pipe, Inches.	Floor Space Required, Inches.
					Strokes.	Gallons.					
000	2½	1½	3	.023	150	3½	¼	⅜	½	⅜	18×4
00	3	1¾	3	.031	150	4½	¼	⅜	¾	½	18×5
0	3¼	2	4	.05	150	7½	½	¾	1¼	1	31×6
1	3½	2¼	4	.07	150	10½	½	¾	1¼	1	29×7
2	4	2½	5	.11	150	16½	½	¾	1¼	1	32×7
3	5	3¼	7	.25	125	31	¾	1	2	1½	44×10
4	5½	3¾	7	.33	125	41	¾	1	2	1½	44×10
4½	7	4	7	.38	125	47	1	1¼	2½	2	45×10
6	7½	5	10	.85	100	85	1	1¼	3	2½	56×16
6½	8	5	12	1.02	100	102	1	1½	4	4	64×16
7	10	6	12	1.47	100	147	1¼	1½	4	4	68×19
8	12	7	12	2.00	100	200	2	2½	5	5	68×19
9	14	8	12	2.61	100	261	2	2½	5	5	68×20
10	16	10	16	5.44	75	408	2½	3	6	6	81×21
.....	16	10	18	6.12	70	428	2½	3	8	6	96×22
.....	18	10	18	6.12	70	428	2½	3	8	6	96×24

The Nos. 0, 1, 2, 3, 4, and 4½ pumps are provided with hand-power attachments. By this means the pump can be used, when steam s down, for filling boilers after "blowing off," washing decks, fire purposes, etc. The hand lever can be easily removed by simply lifting it from the pump.

Nos. 000, 00, and 0 pumps may be built as "reinforced pattern," designed for 400 pounds pressure.

These pumps have large direct water passages and full valve areas, which not only reduce water friction to a minimum, but enable them to be run at a speed that makes them efficient *fire* pumps.

HEAVY PATTERN. FOR 250 POUNDS WATER PRESSURE

Steam Cylinder, Inches.	Water Cylinder, Inches.	Stroke, Inches.	Gallons per Stroke.	Capacity per Minute at Maximum Speed.		Steam Pipe, Inches.	Exhaust Pipe, Inches.	Suction Pipe, Inches.	Delivery Pipe, Inches.	Floor Space Required, Inches.
				Strokes.	Gallons.					
7	$4\frac{1}{2}$	10	.69	100	69	1	$1\frac{1}{4}$	3	$2\frac{1}{2}$	60×14
$7\frac{1}{2}$	5	10	.85	100	85	1	$1\frac{1}{4}$	3	$2\frac{1}{2}$	60×14
8	5	12	1.02	100	102	1	$1\frac{1}{4}$	4	4	66×14
10	6	12	1.47	100	147	$1\frac{1}{4}$	$1\frac{1}{2}$	4	4	69×15
12	7	12	2.00	100	200	2	$2\frac{1}{2}$	5	5	72×20
14	8	12	2.61	100	261	2	$2\frac{1}{2}$	5	5	72×20

Twice the above capacities can be had in emergencies; but for continuous work, such as boiler feeding, about half the speed stated is advised.

The Blake pump (Fig. 162) is operated in a different manner. When the tappet *A* is struck as the piston moves to the left the casting *B* containing the cavities *C*, *D*, and *E* is moved to the left. This motion causes the projection *F* on the casting *B* to cover the passage carrying steam to the right of the auxiliary piston *G* while the projection *H* uncovers the passage leading to the left end of *G*. The projection *I* on the other side of *B* contains two cavities so that this motion to the left covers the passage leading to the left end of the auxiliary piston, while that leading to the right-hand end is connected through one of the cavities under *I* to the exhaust. The steam then entering the left of *G* when the cavity at the right is connected to the exhaust causes the piston to travel to the right and with it the main valve *K*, thus reversing the pump. The action is then repeated in the reverse direction when the arm strikes the tappet *M*.

Should the auxiliary piston fail to operate, the movement of the casting *B* is such that the left-hand passage would be moved to the left so far that steam would be admitted to the

left-hand end of the main cylinder, while exhaust would occur

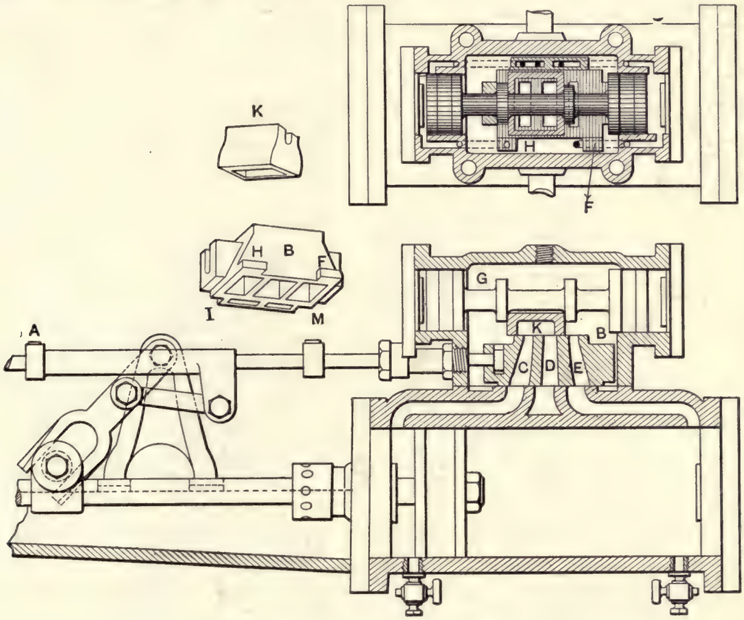


FIG. 162.—Blake Steam Pump.

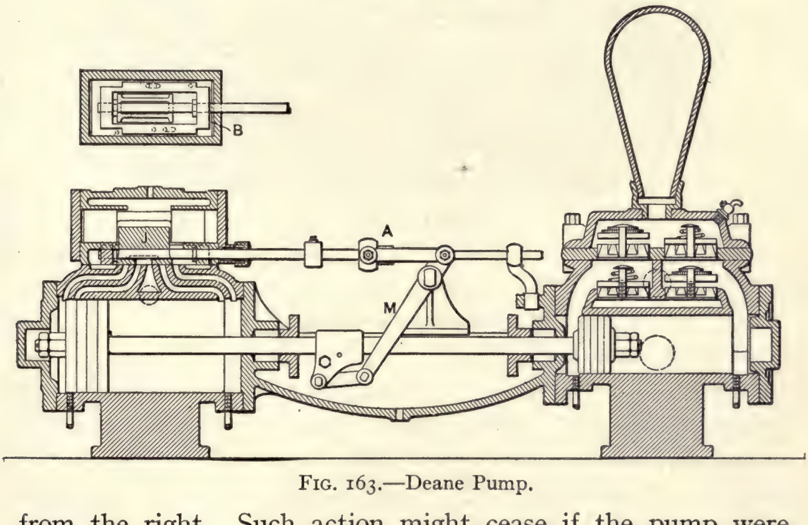


FIG. 163.—Deane Pump.

from the right. Such action might cease if the pump were running slowly.

The passages operating the auxiliary valve are made in the cylinder casting, and although small, they carry sufficient steam to operate the auxiliary piston. The casting *B* is really the valve operating the auxiliary piston. It contains passages leading to the main passage because it is necessary to connect these with the main valve. The same action could be obtained if the valve portion of this was made like a frame surrounding

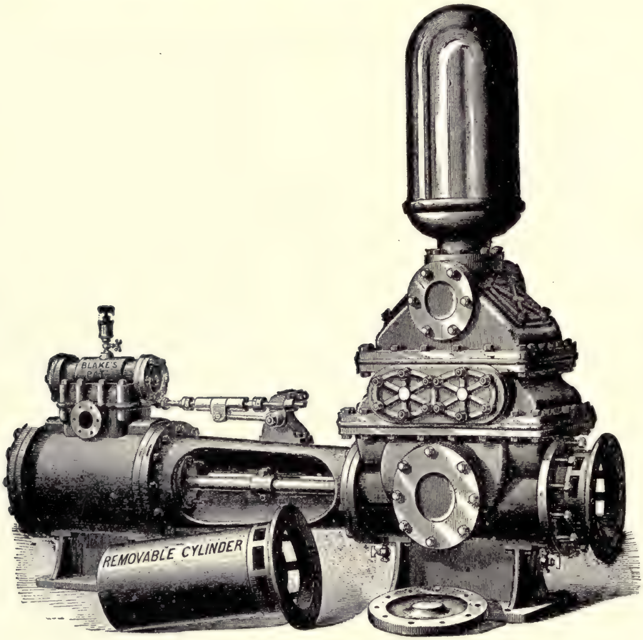


FIG. 164.—Blake Pump.

the main valve while the main valve moved on the lower seat. Such an arrangement is used on the Deane pump.

The Deane pump contains a frame *B* surrounding the main valve *J*. The tappet *A* (Fig. 163) is struck by the sleeve and moves a frame *B* so that the right-hand end of the auxiliary cylinder is connected to the exhaust, while the left end is connected to the steam supply. In this way the auxiliary piston is driven to the right and with it the main valve is carried over, reversing the pump. Should the auxiliary piston cease to act,

the continued motion of the pump would pull the main valve to the right by means of the reverse lever M and thus reverse the pump. The frame B is really a valve operating the auxiliary piston. One end of this controls the admission and exhaust of steam, from one end corresponding to one cylinder end, while the other of the frame controls the other end.

These pumps are accurate in their action. Fig. 164 shows

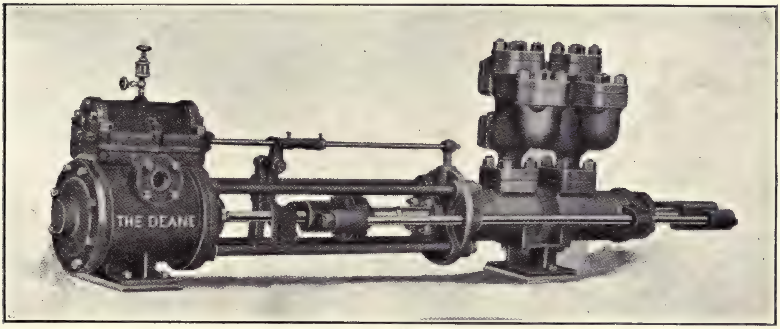


FIG. 165.—Deane Pressure Pump.
(Size $18 \times 2\frac{1}{2} \times 18$.)

the exterior view of the Blake pump and Fig. 165 shows a long-stroke pressure type of the Deane pump, to which the table on p. 180 refers.

THE DEANE SINGLE OUTSIDE-PACKED DOUBLE-PLUNGER PUMP—POT-VALVE PATTERN

For pressures between 300 and 3000 lbs. the style of pump shown in Fig. 165 is recommended. As will be seen, the arrangement of the plungers and cylinders is similar to the regular outside-plunger type, with the difference, however, that the valves are placed in pots above the cylinder. These valves are specially designed for the service; are each in separate compartments, and may be readily inspected by the removal of covers. The cylinders are made of steeline, a special hydraulic cylinder mixture, for all pressures up to and including 1250 lbs. For higher pressures water ends are made of open-hearth steel castings. These pumps are adapted for hydraulic cranes, presses, punches, shears, riveting machines, etc. All parts are

quickly accessible, devoid of complications, and heavy in construction. These pumps are also built with duplex compound and triple-expansion steam ends. Water ends of pumps listed below are of steeline good for a working pressure of 1250 lbs.

Size.			Capacity.			Pipe Sizes.			
Diameter of Steam Cyl.	Diameter of Plunger.	Length of Stroke.	Gallons per Stroke.	Strokes per Minute of One Plunger.	Gallons per Minute of Both Plungers.	Diameter of Steam Pipe.	Diameter of Exh'st Pipe.	Diameter of Suction Pipe.	Diameter of Discharge Pipe.
10	2¼	12	.21	40 to 80	8.5 to 17	1½	2	2	1¼
12	2¼	12	.21	40 to 80	8.5 to 17	2	2½	2	1¼
14	2¼	12	.21	40 to 80	8.5 to 17	2	2½	2	1¼
16	2¼	18	.31	40 to 80	12.5 to 25	2	2½	2	1¼
18	2¼	18	.31	40 to 80	12.5 to 25	3	3½	2	1¼
20	2¼	24	.41	40 to 60	16.5 to 25	3	3½	2	1¼
24	2¼	24	.41	40 to 60	16.5 to 25	4	4½	2	1¼
12	2½	12	.25	40 to 80	10 to 20	2	2½	2	1¼
14	2½	12	.25	40 to 80	10 to 20	2	2½	2	1¼
16	2½	18	.38	40 to 80	15 to 30	2	2½	2	1¼
18	2½	18	.38	40 to 80	15 to 30	3	3½	2	1¼
20	2½	24	.51	40 to 60	20 to 30	3	3½	2	1¼
24	2½	24	.51	40 to 60	20 to 30	4	4½	2	1¼
12	2¾	12	.31	40 to 80	12.5 to 25	2	2½	2	1½
14	2¾	12	.31	40 to 80	12.5 to 25	2	2½	2	1½
16	2¾	18	.46	40 to 80	18.5 to 37	2	2½	2	1½
18	2¾	18	.46	40 to 80	18.5 to 37	3	3½	2	1½
20	2¾	24	.62	40 to 60	25 to 37	3	3½	2	1½
24	2¾	24	.62	40 to 60	25 to 37	4	4½	2	1½
14	3	12	.37	40 to 80	15 to 30	2	2½	2	1½
16	3	18	.55	40 to 80	22 to 44	2	2½	2	1½
18	3	18	.55	40 to 80	22 to 44	3	3½	2	1½
20	3	24	.73	40 to 60	29 to 48	3	3½	2	1½
24	3	24	.73	40 to 60	29 to 48	4	4½	2	1½
16	3½	18	.75	40 to 80	30 to 60	2	2½	3	2
18	3½	18	.75	40 to 80	30 to 60	3	3½	3	2
20	3½	24	1.00	40 to 60	40 to 60	3	3½	3	2
24	3½	24	1.00	40 to 60	40 to 60	4	4½	3	2
18	4	18	.98	40 to 80	39 to 78	3	3½	3½	2½
20	4	24	1.30	40 to 60	52 to 104	3	3½	3½	2½
24	4	24	1.30	40 to 60	52 to 104	4	4½	3½	2½

Pumps of this type are made for heavier pressures.

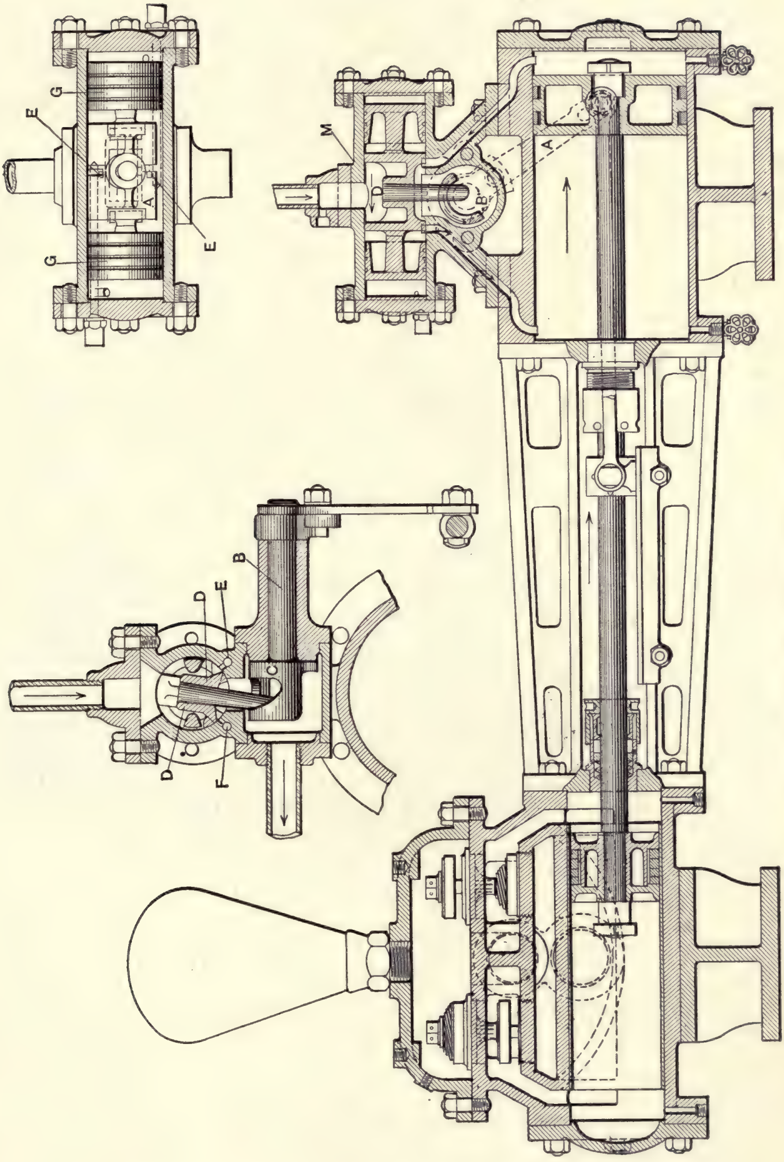


Fig. 166.—Davidson Steam Pump.

The Davidson pump (Fig. 166) is operated by means of the oscillation of the auxiliary valve. The arm *A* is attached to the piston rod on the outside of the pump. The turning

of the shaft B turns the cam C which causes the auxiliary valve D to turn. When the piston gets to the end of the stroke the pin puts the valve into the position shown so that the passage E is connected with the steam and the passage F with the exhaust. This causes the auxiliary pistons GG to be driven to the left so that steam will enter the right end of the main cylinder. The auxiliary valve is oscillated when it acts as an auxiliary valve, but the same casting D acts as the main valve by its longitudinal motion. The casting D fits into a

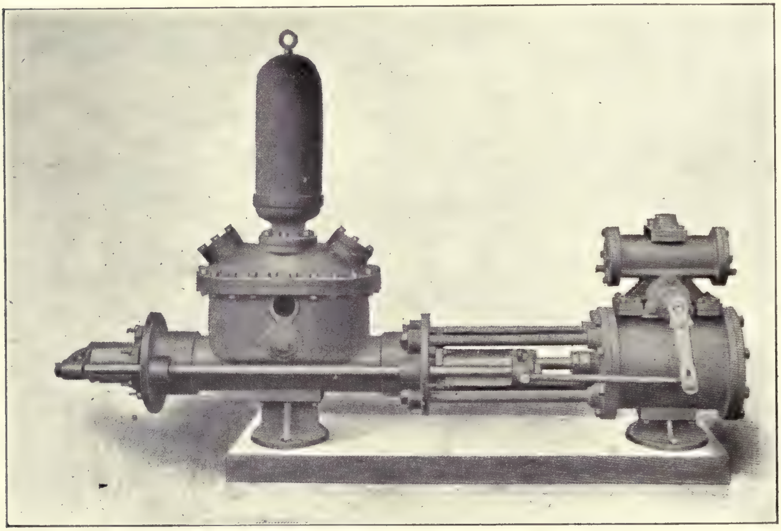


FIG. 167.—Davidson Pump.

cavity in the frame of the auxiliary pistons GG, and although it is carried to the top of the cylinder in which GG moves, it does not completely fill the cylinder, so that the steam which enters at H can pass around the valve D. The exhaust passes through the cavity containing the pin of D and the cam C.

The valve D is twisted into the position shown in the small section whenever it is preparing to drive the main valve to one end of the cylinder. When it is in such a position that E and F are covered, the openings to the main cylinder are open to either steam or exhaust. In this way, when steam is turned

THE DAVIDSON PRESSURE PUMP, PISTON PATTERN

For Boiler Feeding, Operating Hydraulic Elevators, etc.

For a pressure of 150 pounds. (If specially ordered these pumps can be furnished for 250 pounds working pressure).

(Composition or bronze cylinders to order.)

SIZES AND DETAILS

Size. No.	Steam Cylinder.	Water Cylinder.	Stroke Inches	Gallons per Stroke.	H.P Boiler, based on 30 lbs. of Water per H.P.perHr. which the Pump will supply with ease.[1]	Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Discharge Pipe.
0	2½	1½	3	.022	20	¼	⅜	¾	½
½	3	1¾	4	.041	35	¼	⅜	1	¾
1	3½	2	4	.05	50	⅜	½	1¼	1
1½	4	2¼	4	.069	65	⅜	½	1¼	1
2	4½	2½	6	.13	125	½	¾	1½	1¼
2½	5	3	6	.183	175	½	¾	2	1½
3	5½	3¼	8	.28	275	¾	1	2	1½
3½	6	4	8	.435	425	¾	1	2½	2
4	7	4	10	.54	525	¾	1	2½	2
4½	8	5	10	.85	850	1	1¼	3	2½
5	9	5¼	12	1.12	1100	1	1¼	3	2½
6	10	6	12	1.47	1400	1	1½	4	3
7	12	7	12	2.00	2000	1½	2	5	4
7½	14	8¼	12	2.77	2750	1½	2	6	5
8	14	8¼	14	3.23	3200	1½	2	6	5
8½	14	10	14	4.76	4750	1½	2	8	7
9	16	9¼	16	4.66	....	2	2½	7	6
9½	16	10	16	5.44	....	2	2½	8	7
10	18	10½	18	6.74	....	2½	3	8	7
10½	18	11½	18	8.09	....	2½	3	9	8
11	20	11½	20	9.00	....	2½	3	9	8
11½	20	13	20	11.49	....	2½	3	10	9
12	22	13	22	12.64	....	2½	3	10	9
13	24	14	24	16.00	....	3½	4	10	10

[1] Capacities for boiler feeding based on a speed of 60 single strokes per minute with feed water at ordinary temperature. With high temperature feed water reduce H.P. rating one-third. These pumps can be run as slowly as may be desired, and in cases of emergency, speeded up to about twice ordinary capacities.

Suction and discharge openings on both sides. Hand levers furnished with sizes No. 0 to No. 3½, and with No. 4 when ordered. Water-piston packing and valves for hot or cold water as ordered. Every machine thoroughly tested before leaving works.

on either steam will enter E or F and drive the main valve to a position to admit steam to one end and start the pump, or the main valve itself will be in a position to start the pump. From this it may be seen that there is no dead center possible with this pump.

It is claimed that the action of the cam prevents the piston

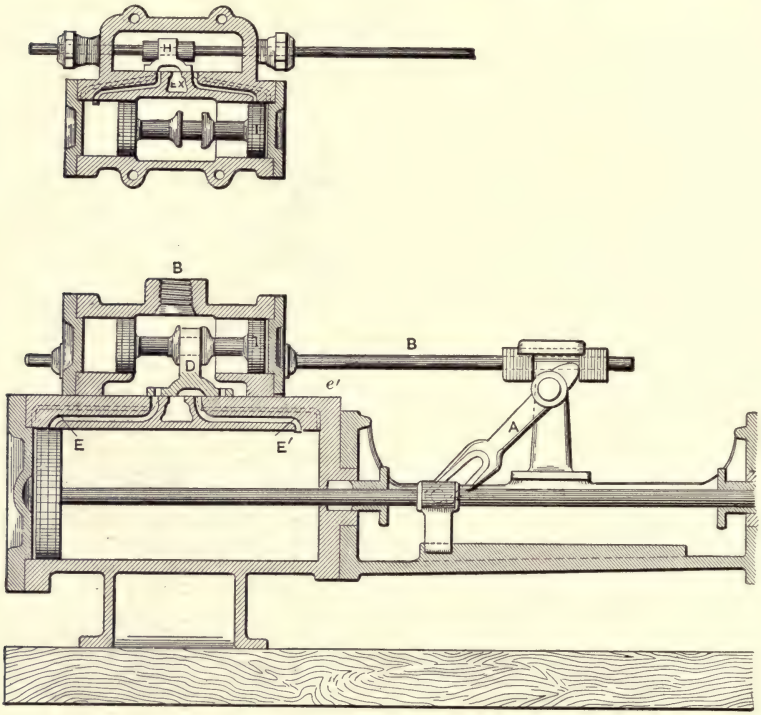


FIG. 168.—Burnham Steam Pump.

from striking the cylinder head. Moreover, the pump should start from any position and make its full stroke.

Fig. 167 gives an outside view of this pump, showing the method of driving the cam lever.

The table on the preceding page gives sizes used by the M. T. Davidson Co. for one type of their pump.

In the Burnham steam pump (Fig. 168) the auxiliary valve is placed on the side of the pump and is driven by the cam rod

A. When the pump reaches the position shown in the figure, the rod *B* is moved to the right and this moves the auxiliary valve *H* to the right. This valve is mounted on the side of the steam chest and its motion to the right will admit steam to the left-hand end of the auxiliary piston *I* while it connects the right-hand end to the exhaust. This drives the auxiliary piston to the position shown in the figure, and with it the main valve. The pump is then reversed.

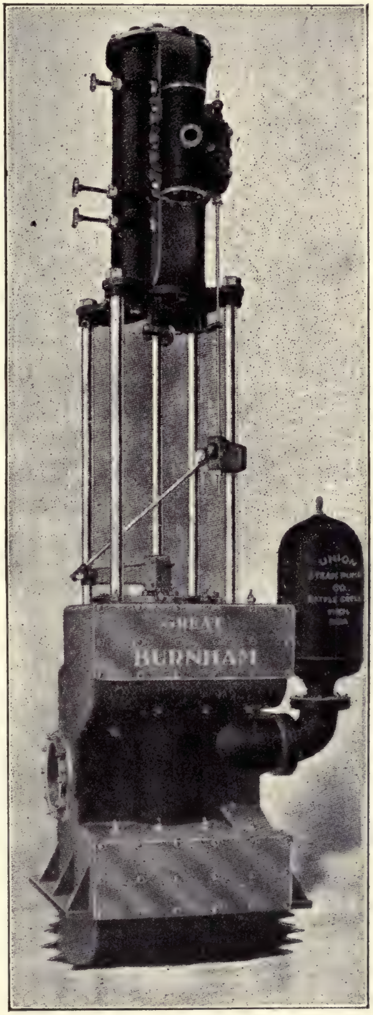


FIG. 169.—Burnham Pump.

The double ports for the main piston and auxiliary piston are arranged to prevent pounding and give smooth action. The small ports leading to the ends of the cylinders are for the sole purpose of admitting steam. The exhaust takes place through the passages shown full in the section and when either the main piston or the auxiliary piston travel over these ports the steam behind them is trapped and forms a cushion. In the positions shown in the figure each of the pistons is started on its return stroke by the small quantities of steam which are admitted through the small passages shown dotted and extending to the ends of the cylinders. Fig. 169 illustrates the exterior appearance of the pump. It will be

noted that the valve cam moves the valve rod at the end of the stroke only, and moreover this motion is gradual and slow.

The table gives the sizes of these pumps used for tank service.

DETAILS OF BURNHAM TANK OR LIGHT-SERVICE PUMPS

Steam Cylinder.	Water Cylinder.	Length of Stroke.	As length of connections increases, larger size pipes should be used.				Gals. per Stroke.	Gallons per Minute at Given Number of Strokes.	Gallons per Hour at Highest Speed.	Strokes per Minute Varying with Kind of Work and Pressure.	Lining.
			Steam.	Exh'st.	Suction.	Disch'ge.					
8	10	12	1	1½	8	6	4.08	204 to 408	24,400	50 to 100	R
8	11	12	1	1½	8	6	4.93	247 to 494	29,600	50 to 100	R
8	12	12	1	1½	8	6	5.87	293 to 587	35,200	50 to 100	R
8½	6	10	1	1½	4	3½	1.22	73 to 147	8,800	60 to 120	R
8½	7	10	1	1½	5	4	1.66	100 to 200	12,000	60 to 120	R
8½	8	10	1	1½	5	4	2.17	130 to 261	15,600	60 to 120	R
8½	9	10	1	1½	6	5	2.75	165 to 330	19,800	60 to 120	R
8½	10	10	1	1½	6	5	3.40	204 to 408	24,400	60 to 120	R
10	8	12	1½	2½	5	4	2.61	130 to 261	15,600	60 to 120	R
10	9	12	1½	2½	6	5	3.30	165 to 330	19,800	50 to 100	R
10	10	12	1½	2½	8	6	4.08	204 to 408	24,400	50 to 100	R
10	11	12	1½	2½	8	6	4.93	247 to 494	29,600	50 to 100	R
10	12	12	1½	2½	8	6	5.87	293 to 587	35,200	50 to 100	R
10	14	16	1½	2½	10	8	10.65	404 to 808	48,400	38 to 75	R
12	8	12	1½	2½	5	4	2.61	130 to 261	15,600	50 to 100	R
12	9	12	1½	2½	6	5	3.30	165 to 330	19,800	50 to 100	R
12	10	12	1½	2½	8	6	4.08	204 to 408	24,400	50 to 100	R
12	11	12	1½	2½	8	6	4.93	247 to 494	29,600	50 to 100	R
12	12	12	1½	2½	8	6	5.87	293 to 587	35,200	50 to 100	R
12	14	12	1½	2½	10	8	8.00	400 to 800	48,000	50 to 100	R
12	8	16	1½	2½	6	4	3.48	132 to 264	15,800	38 to 75	R
12	9	16	1½	2½	6	5	4.40	167 to 334	20,040	38 to 75	R
12	10	16	1½	2½	8	6	5.44	206 to 412	24,700	38 to 75	R
12	11	16	1½	2½	8	6	6.58	250 to 500	30,000	38 to 75	R
12	12	16	1½	2½	8	6	7.82	297 to 594	35,600	38 to 75	R
12	14	16	1½	2½	10	8	10.65	404 to 808	48,400	38 to 75	R
14	10	12	2	2½	8	6	4.08	204 to 408	24,400	50 to 100	R
14	12	12	2	2½	8	6	5.87	293 to 587	35,200	50 to 100	R
14	14	12	2	2½	10	8	8.00	400 to 800	48,000	50 to 100	R
14	10	16	2	2½	8	6	5.44	206 to 412	24,700	38 to 75	R
14	11	16	2	2½	8	6	6.58	250 to 500	30,000	38 to 75	R
14	12	16	2	2½	8	6	7.82	297 to 594	35,600	38 to 75	R
14	14	16	2	2½	10	8	10.65	404 to 808	48,400	38 to 75	R
14	16	16	2	2½	10	8	14.00	532 to 1064	63,800	38 to 75	R
14	16	20	2	2½	12	10	17.40	522 to 1044	62,600	30 to 60	R
14	18	20	2	2½	12	10	22.03	660 to 1321	79,200	30 to 60	R
16	11	16	2	2½	8	6	6.58	250 to 494	29,600	38 to 75	R
16	12	16	2	2½	8	6	7.82	297 to 594	35,600	38 to 75	R
16	14	16	2	2½	10	8	10.65	404 to 808	48,400	38 to 75	R
16	16	16	2	2½	10	8	14.00	532 to 1064	63,800	38 to 75	R
16	16	20	2	2½	12	10	17.40	522 to 1044	62,600	30 to 60	R
16	18	20	2	2½	12	10	22.03	660 to 1321	79,200	30 to 60	R
16	20	24	2	2½	14	12	32.63	816 to 1632	97,920	25 to 50	R
16	22	24	2	2½	16	12	39.50	987 to 1975	118,500	25 to 50	R
18	14	16	2½	3½	10	8	10.65	405 to 810	48,600	38 to 75	R
18	16	20	2½	3½	12	10	17.40	522 to 1044	62,600	30 to 60	R
18	18	20	2½	3½	12	10	22.03	660 to 1321	79,200	30 to 60	R
20	16	20	2½	3½	12	10	17.40	522 to 1044	62,600	30 to 60	R
20	18	20	2½	3½	12	10	22.03	660 to 1321	79,200	30 to 60	R
20	20	24	2½	3½	14	12	32.63	816 to 1632	97,920	25 to 50	R
20	22	24	2½	3½	16	12	39.50	987 to 1975	118,500	25 to 50	R

"R" signifies *removable* bronze lining in water cylinder. "P" signifies bronze lining *pressed* in.

The Dean Brothers pump (Fig. 170) illustrates another method of obtaining these results. As in the Burnham pump the auxiliary valve is placed on the side of the main steam chest. This pump differs from the others considered in that the motion of the auxiliary valve is continuous and not intermittent. This valve is driven from a reverse lever attached by a link to the piston rod, and the valve is moved by a link at the upper end of this. When the main piston reaches the extreme right of its stroke the auxiliary valve is at its extreme left, and the edge of the valve uncovers a small circular port of the passage

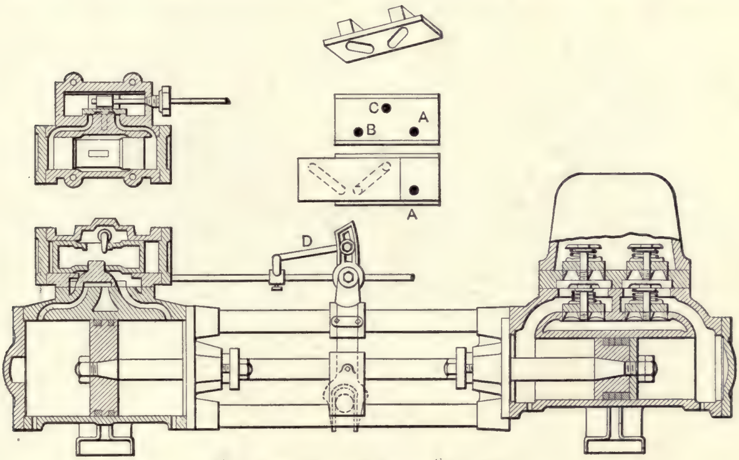


FIG. 170.—Dean Brothers Pump.

A leading to the right end of the auxiliary cylinder. At this time one of the two diagonal grooves on the under side of the valve connects the other passage *B* to the exhaust port *C*. This then forces the auxiliary piston to its extreme left, and with it the main *D* slide valve, admitting steam to the right-hand end of the cylinder, reversing the motion. The auxiliary valve begins to move to the right, cutting off the steam and exhaust from the auxiliary valve. This condition does not change until the valve almost reaches the end of its stroke, when the other diagonal groove connects *A* to *C*, while *B* is uncovered to the action of the live steam. The auxiliary valve then goes over to the other position and the

pump reverses. The link D, which moves the auxiliary valve rod, may be moved in a slot closer to or farther from the pivot of the reverse lever, making its stroke shorter or longer; this makes the stroke of the main pump longer or shorter, as it means more or less motion of the main piston to move the auxiliary valve a sufficient distance to bring the ports into the position for shifting the auxiliary piston.

The small spindle projecting from the top of the main valve casing is used to shift the auxiliary piston and with it the main valve which rests in a slot in the auxiliary piston. Such a contrivance is necessary, as the main valve may be left in a

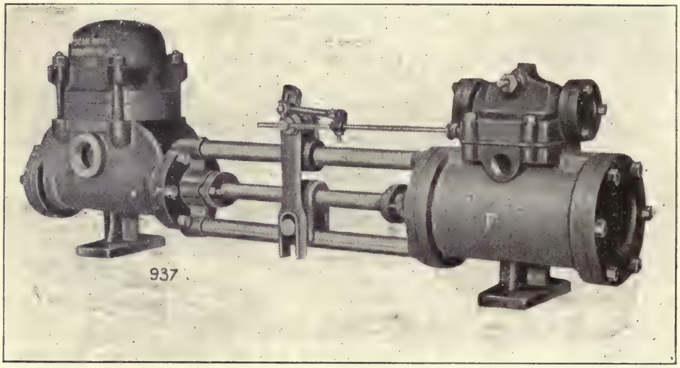


FIG. 171.—Dean Brothers Pump.

position so that steam could not enter either side. This condition rarely occurs, but in most of these pumps such a contingency is guarded against by some device like this or the projecting rods of the Marsh pump.

Fig. 171 illustrates one of these pumps designed for boiler feeding, in which several special features can be seen. The steam and water valves are easily examined without the necessity of interfering with any pipes; the frame is of steel; the cylinders are removable and may be replaced without disturbing any other parts; the piston rod has a cross-head fitting over the lower rod frame, making it impossible to twist the cross-head so as to jam the valve gear.

The following table gives the sizes used by these makers for one line of their pumps.

DEAN BROTHERS SIMPLE PUMPS WITH PACKED PISTONS

(For feeding boilers or pumping against pressure. They will elevate water 200 feet with 60 pounds steam.)

TABLE OF DIMENSIONS

Size.	Steam, Cylinder. Inches.	Water Cylinder. Inches.	Stroke, Inches.	Gallons, per Stroke.	[1] Capacity per Minute. Strokes.	Gallons.	Steam Pipe Inches.	Exhaust Pipe, Inches.	Suction Pipe, Inches.	Delivery Pipe, Inches.	Limit of Capacities for Feeding Boilers, H.P.
C	3	2	4	0.054	200	10	$\frac{3}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	1	50
D	4	$2\frac{5}{8}$	6	0.14	140	20	$\frac{1}{2}$	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	100
E	$5\frac{1}{2}$	$3\frac{3}{4}$	7	.33	125	42	$\frac{3}{4}$	1	2	$1\frac{1}{2}$	220
F	6	4	10	.55	125	68	1	$1\frac{1}{4}$	3	$2\frac{1}{2}$	400
G	8	5	12	1.02	100	102	1	$1\frac{1}{2}$	4	3	700
H	9	6	12	1.47	100	147	$1\frac{1}{4}$	2	4	3	1000
I	10	7	12	2.00	100	200	$1\frac{1}{4}$	2	5	4	1300
K	12	8	12	2.61	100	261	2	$2\frac{1}{2}$	6	5	1600
L	14	9	16	4.40	80	352	2	3	7	6	2500
M	14	10	20	6.80	70	476	2	3	8	7	3500

[1] In an emergency more than the above capacity can be had, but for continuous work, such as feeding boilers, not more than half the strokes given above are advised. The valve motion secures a smooth action and admits of regulation, so as to deliver a steady supply of water, exactly equal to the amount evaporated.

The pumps described in this chapter are taken to represent this class, and although there are many other different forms, the principles shown in the these will aid in understanding the action of any other simplex pump. All of the valve gears aim to move the valve positively whatever be its speed. From the spring-thrown valve of Worthington to the latest form, this is the governing idea.

CHAPTER V

DYNAMICS OF WATER END

To make the velocity of discharge from a pump more definite a pump with a fly wheel and connecting rod will be considered. The figure below (Fig. 172) represents a plunger pump in which the rate of discharge or suction at any instant is proportional to the velocity of the plunger. From the diagram the movement from the end of the stroke is

$$x = AB + BC = r(1-\cos\theta) + nr(1-\cos\alpha), \quad . \quad . \quad (1)$$

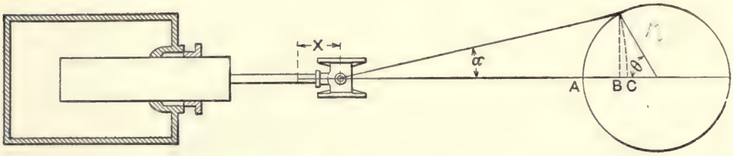


FIG. 172.—Diagram of Pump.

where θ = the angle the crank has moved from its head dead center;

α = the inclination of the connecting rod with the center line;

r = the crank radius in feet;

n = the ratio of length of connecting rod to crank radius.

Now

$$\sin\alpha = \frac{r}{nr}\sin\theta.$$

Hence

$$\cos\alpha = \sqrt{1 - \frac{1}{n^2}\sin^2\theta} = 1 - \tfrac{1}{2}\frac{1}{n^2}\sin^2\theta, \text{ approx.}$$

Therefore

$$x = r\left[1 - \cos\theta + \tfrac{1}{2}\frac{1}{n}\sin^2\theta\right]. \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\text{Velocity}=\frac{dx}{dt}=S=r\left[\sin\theta+\frac{\sin\theta\cos\theta}{n}\right]\omega \quad . \quad . \quad (3)$$

$$=r\left[\sin\theta+\frac{1}{2n}\sin 2\theta\right]\omega.$$

$$\text{Acceleration}=\frac{d^2x}{dt^2}=a=r\left[\cos\theta+\frac{\cos 2\theta}{n}\right]\omega^2. \quad . \quad . \quad . \quad (4)$$

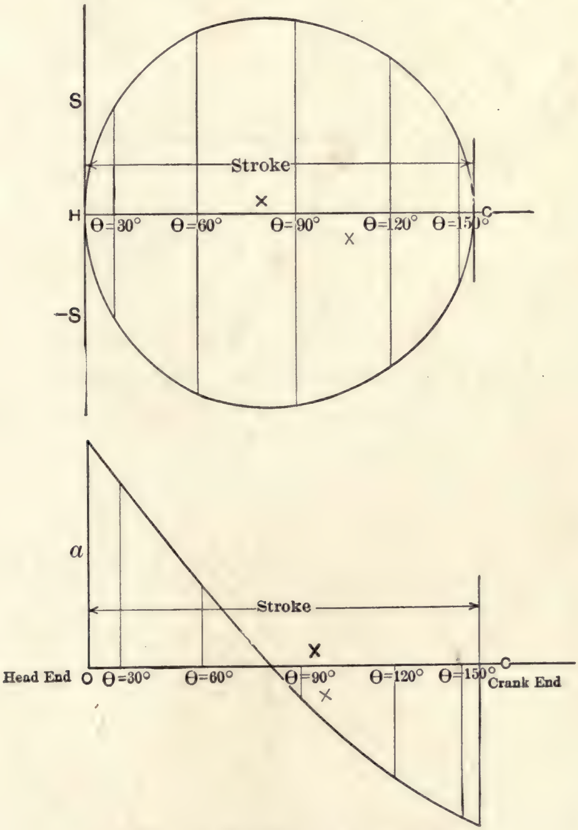


FIG. 173.—Velocity and Acceleration.

These give diagrams such as shown in Fig. 173, when S and a are plotted against x or piston position as asbcissæ.

This approximate solution is sufficiently close for general work, as may be seen from the table below, which gives the

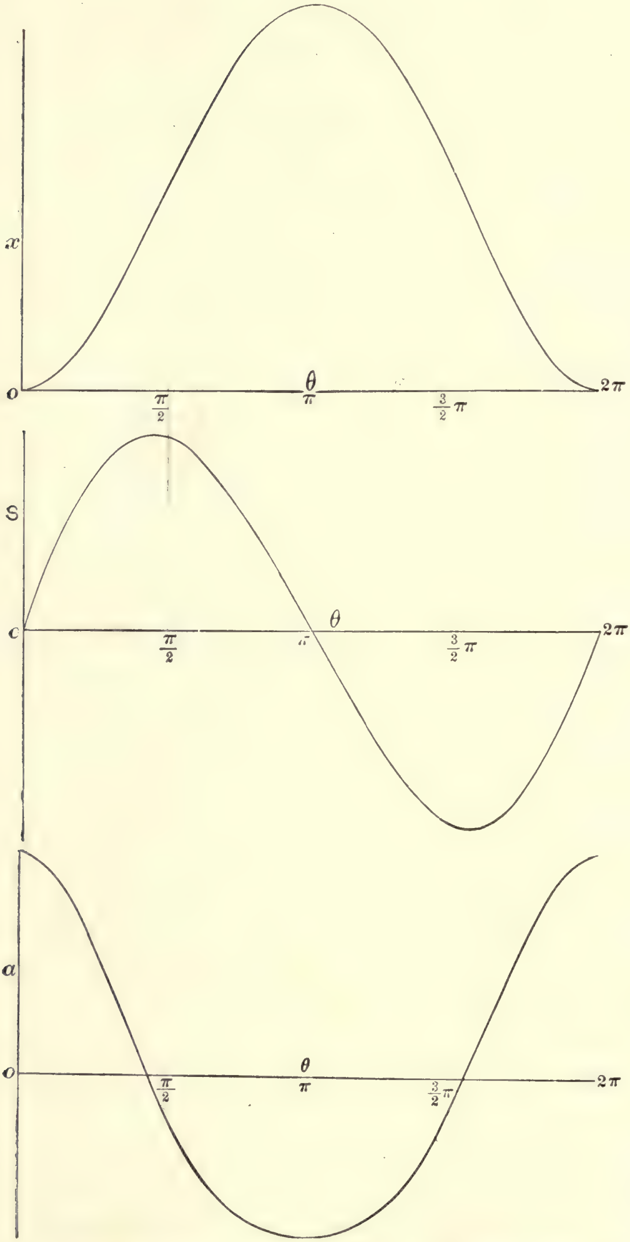


FIG. 174.—Curves of x, S, a for Different Crank Angles.

values of a by the approximate formula and the exact formula obtained without expanding the equation for cos α. These exact equations are

$$S=\omega r\left[\sin\theta+\frac{\sin\theta\cos\theta}{\sqrt{n^2-\sin^2\theta}}\right]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$a=\omega^2 r\left[\cos\theta+\frac{n^2\cos^2\theta}{(n^2-\sin^2\theta)^{\frac{3}{2}}}-\frac{\sin^2\theta}{(n^2-\sin^2\theta)^{\frac{1}{2}}}\right]. \quad . \quad (6)$$

For a discussion of the use of the exact and approximate values of acceleration the reader is referred to the papers of D. S. Jacobus, A. S. M. E. Transactions, Vol. XI, p. 492 et seq., and p. 1116 et seq.

In the exact formula the angular velocity is considered as constant, and since this is not the case there is really an error to the exact formula.

The curves, Fig. 174, give the values of x, S, and a as functions of crank movement. These curves are symmetrical, and in computing the values of the points, as in the table below, for a unit value of ω and r, this symmetry is seen.

θ	cos θ	sin θ	cos 2θ	sin 2θ	x	S	a	a exact
0	1	0	1	0	0	0	1.167	1.167
15	.96593	.25882	.86603	.5	.040	.300	1.110	1.111
30	.86603	.5	.5	.86603	.155	.572	.949	.950
45	.70711	.70711	0	1	.335	.790	.707	.706
60	.5	.86603	−.5	.86603	.563	.938	.417	.417
75	.25882	.96593	−.86603	.5	.819	1.007	.114	.113
90	0	1	−1	0	1.083	1.000	−.167	−.169
105	−.25882	.96593	−.86603	−.5	1.346	.924	−.403	−.405
120	−.5	.86603	−5	−.86603	1.562	.794	−.583	−.583
135	−.70711	.70711	0	−1	1.749	.624	−.707	−.709
150	−.86603	.5	.5	−.86603	1.908	.428	−.783	−.782
165	−.96593	.25882	.86603	−.5	1.972	.217	−.822	−.821
180	−1	0	1	0	2	0	−.833	−.833

To get actual S the values are multiplied by $r\omega$, a by $r\omega^2$, and x by r.

The momentary rate of discharge from any pump will be $A\times$velocity, where A represents the effective area of the piston or plunger. A has various values for the different types of

pumps. If D = diameter of cylinder and d = diameter of rod, the following results:

CASE 1. For the plunger pump, $A = \dfrac{\pi D^2}{4}$.

CASE 2. $\begin{cases} \text{For the piston pump, on end without rod, } \dfrac{\pi D^2}{4}. \\ \text{For the piston pump, on end with rod, } \dfrac{\pi}{4}\left(D^2 - d^2\right). \end{cases}$

CASE 3. $\begin{cases} \text{For the bucket pump down stroke, } \dfrac{\pi d^2}{4}. \\ \text{For the bucket pump up stroke, } \dfrac{\pi}{4}\left(D^2 - d^2\right). \end{cases}$

Thames-Ditton pump down stroke, $\dfrac{\pi d^2}{4}$.

Thames-Ditton pump up stroke, $\dfrac{\pi}{4}\left(D^2 - d^2\right)$.

The discharge then becomes

$$\text{Total } Q = \int dQ = \int_0^t A \frac{dx}{dt} dt, \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$Q = A \int_0^x dx = A \times \text{distance passed over,}$$

$$= AlN'. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

l = length of stroke,
N' = effective pumping strokes in given time.

The action of the discharge or suction may be shown by a diagram, Fig. 175, in which abscissæ represent time and ordinates, the quantity $A\dfrac{dx}{dt}$. The area of this curve is Q and the mean height of the curve would be

$$\frac{\text{total } Q}{t} = \frac{AlN'}{t} = AlN''. \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

N'' = effective strokes per second.

To show the variation of the momentary values from the

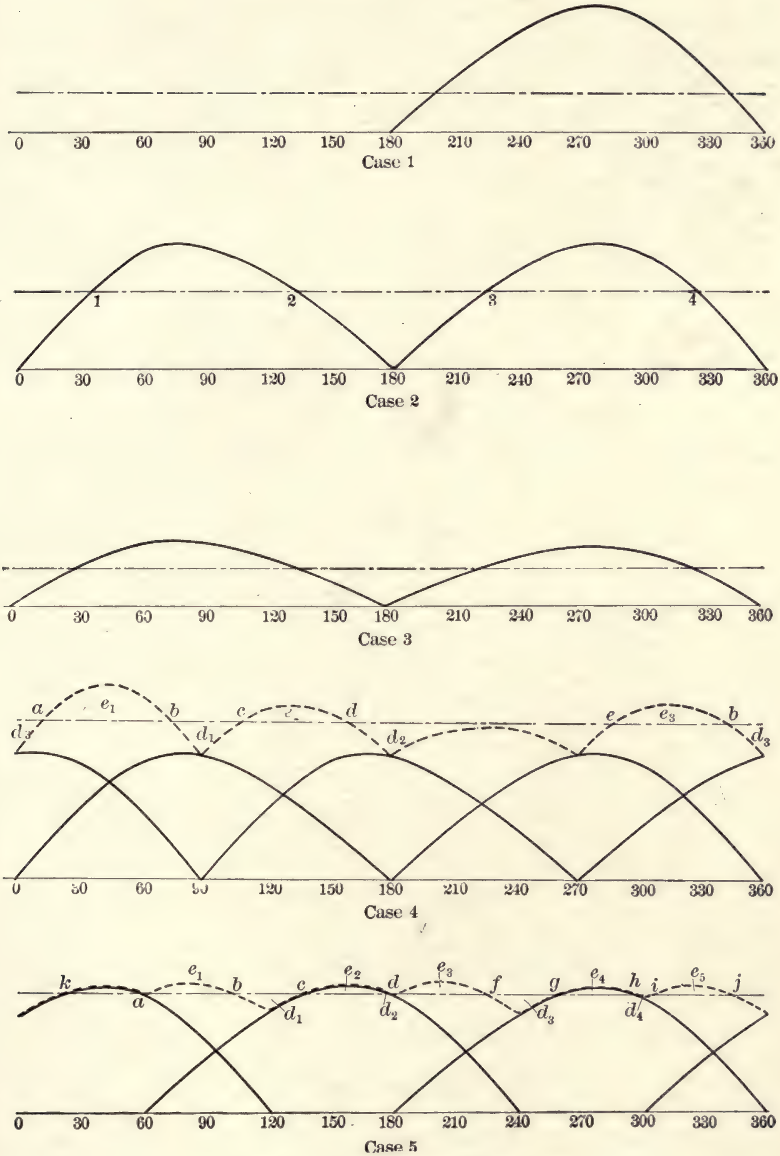


FIG. 175.—Pump Discharges.

mean, the mean value may be plotted (Fig. 175) as a dot and dash line on the diagram. Two additional cases have been

added: One, Case 4, in which there are two double-acting pumps and another, Case 5, where three single-acting pumps are used.

The series of diagrams are drawn for pumps in which the diameter of the pump barrel is 20 inches; the diameter of the piston rod, where used, 2 inches; the diameter of the equalizing plunger rod, where used, 14 inches, and the stroke 30 inches. The speed of the pumps will be taken at sixty revolutions per minute and $n=6$. The quantities discharged per second for the various cases are as follows:

CASE 1. $$Q=\frac{\pi D^2}{4}\times L\times N''=\frac{314.16}{144}\times\frac{30}{12}\times 1=5.4475.$$

CASE 2. $$Q=\left[\frac{\pi D^2}{4}+\frac{\pi}{4}(D^2-d^2)\right]\times L\times N''$$

$$=\left[\frac{314.16}{144}+\frac{311.02}{144}\right]\times\frac{30}{12}\times 1=10.8475.$$

$$A_h=\frac{\pi D^2}{4}=\frac{314.16}{144}=2.181.$$

$$A_c=\frac{\pi}{4}(D^2-d^2)=\frac{311.02}{144}=2.160.$$

CASE 3. $$Q=\frac{\pi D^2}{4}\times L\times N''=\frac{314.16}{144}\times\frac{30}{12}\times 1=5.4475.$$

$$A_{\text{down}}=\frac{\pi d^2}{4}=\frac{153.94}{144}=1.069.$$

$$A_{\text{up}}=\frac{\pi}{4}(D^2-d^2)=\frac{160.22}{144}=1.112.$$

CASE 4. $Q=2Q$ of Case 2.

A_h and A_c same as Case 2.

CASE 5. $Q=3Q$ of Case 1.

$A=$ same as Case 1.

The curves show clearly the way the quantity varies as the pump moves. The variation due to the piston rod is

shown in the figure by the different heights of the curve on the forward and back stroke of the double-acting pump. The effect of a number of cylinders on the uniformity of flow in the discharge pipe is seen by comparing the curve from the

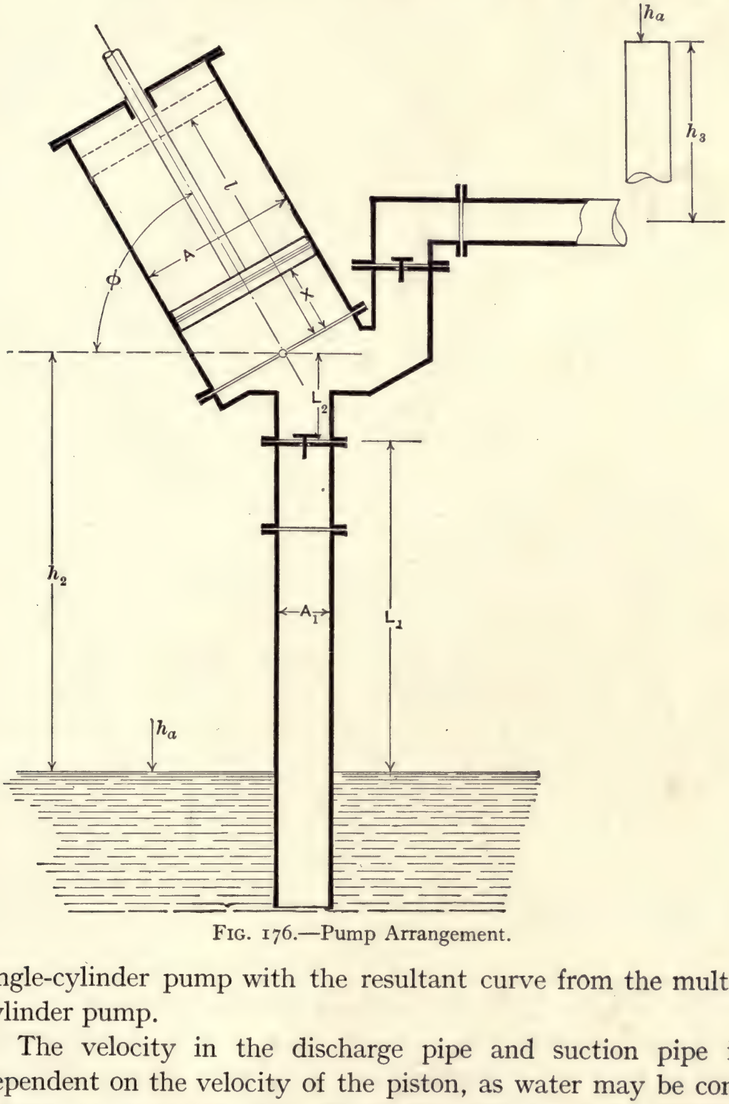


FIG. 176.—Pump Arrangement.

single-cylinder pump with the resultant curve from the multi-cylinder pump.

The velocity in the discharge pipe and suction pipe is dependent on the velocity of the piston, as water may be considered as incompressible. Since this will mean a variation

in pressure due to the change of momentum, it is necessary to consider the forces acting in the suction pipe and delivery pipe.

Consider one side of the pump, Fig. 176, having a piston area A and a stroke l. To make this a general case the cylinder will be inclined at an angle ϕ to the horizontal. The dimensions L represent lengths, while $A'A$ are the areas of section of the various channels through which the water passes, and h, the pressures measured in feet of water at the various points.

The first pressure necessary to find is that acting on the lower side of the piston during an upward stroke. This may be written as

$$h_s = h_a - (h_2 + h_3 + h_4 + x \sin \phi), \quad . \quad . \quad . \quad (10)$$

h_s = pressure on suction side of piston expressed in feet;
h_a = head corresponding to the pressure of the atmosphere;
h_2 = height to end of piston stroke from water level;
h_3 = friction and other losses except that of inertia;
h_4 = force due to inertia expressed in feet head;
$x \sin \phi$ = the vertical movement of the piston.

The pressure h_s is the mean pressure on the piston and the actual pressure at the center of the piston; there is a slight variation over the area.

The friction losses in the suction side are made up of three principal parts: That in the suction pipe, that in the valve boxes, and that in the passages of the pump. The general formula for this loss at any instant is

$$h = f \frac{lv^2}{d2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

or using later experimental work,

$$h = klv^n. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The velocity of the water in any member of the pump varies in the same manner as that of the piston provided the water stream does not separate but follows the piston. Hence a mean value of the loss must be obtained.

Since v^2 varies, the mean loss per cu.ft. is

$$h_{fs}=\frac{\int_0^Q f\frac{l_1}{d_1}\frac{v_1^2}{2g}dq}{Q}=\frac{\int_0^1 f\frac{l_1}{d_1}\frac{v_1^2}{2g}AS\,dt}{Q}.$$

since $dq=AS\,dt$, where $S=$the speed of the piston.

Now $V_1=\frac{AS}{A_1}$. This gives

$$h_{fs}=\frac{Al_1}{Q2gd_1}\int_0^1 f\left(\frac{AS}{A_1}\right)^2 Sdt=\frac{A^3l_1}{Q2A_1^2gd_1}\int_0^1 fS^3dt. \quad (13)$$

To integrate this for any given pipe the values of the coeffi-

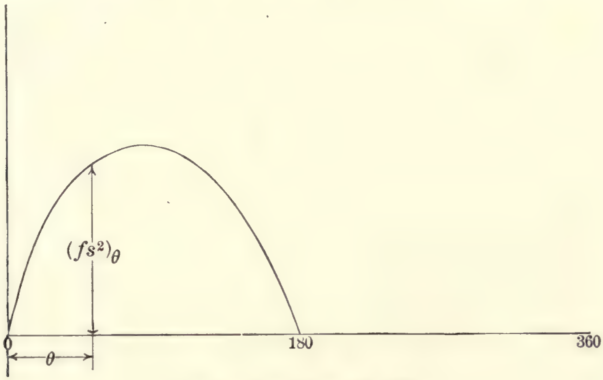


FIG. 177.—Curve of fS^3.

cient f for different velocities V_1 in the pipe are found, and the product fs^3 is found at different intervals of time, or what is the same thing, for different values of θ and are plotted on a θ base. This gives a diagram shown in Fig. 177.

The area of this figure, found by a planimeter or calculated by Simpson's Rules, is the value of the integral and from this the mean h loss may be found.

The instantaneous value of the loss at any point per pound reduces to

$$f\frac{l}{d}\left(\frac{A}{A_1}\right)^2\frac{S^2}{2g}=h=\zeta_s\left(\frac{A}{A_1}\right)^2\frac{S^2}{2g}, \quad . \quad . \quad . \quad . \quad (14)$$

This form of an expression holds for the loss in the passages leading from the valve to the cylinder and through the valve, although there is also a direct loss due to the pressure required to hold up the suction valves. The form will also represent the losses at entrance to the pipe, and other obstruction.

Another loss to be considered is the loss due to sudden changes in velocity along the water line due to changes in the section. These losses are of the form,

$$h=\frac{(V_1-V_2)^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

The losses of pressure include those of velocity heads, and since the water has finally a velocity S in the pump, the velocity head, $\frac{S^2}{2g}$, must be one of the terms of the reduction in pressure, as there was originally no velocity in the fore bay or place from which the water is drawn. The initial velocity, $\frac{AS}{A_1}$, in the suction pipe is equivalent to, and requires a pressure, $\left(\frac{AS}{A_1}\right)^2\frac{1}{2g}$, which is greater than $\frac{S^2}{2g}$. This greater velocity head, however, is changed into pressure head as the velocity decreases, so that when the piston is reached the only part of velocity head left which requires a pressure from the piston is $\frac{S^2}{2g}$.

Combining these, there results:

$$h_3=\Sigma\zeta_s\left(\frac{A}{A_1}\right)^2\frac{S^2}{2g}+h_{sv}+\Sigma\zeta'_s\left(\frac{A}{A_1}\right)^2\frac{S^2}{2g}+\frac{S^2}{2g}$$

$$=\Sigma\zeta_s\left(\frac{A}{A_1}\right)^2\frac{S^2}{2g}+h_{sv}+\frac{S^2}{2g}. \quad . \quad (16)$$

To get this term the value of ζ must be obtained from tables and h_{sv} is obtained, as described later.

The loss h_4 is computed by considering the inertia of the water.

The masses of water in the various parts of the pump are as follows:

In the pump AXw in pounds.
In the pump spaces A_2L_2w in pounds.
In suction pipe A_1L_1w in pounds.

These masses when multiplied by the acceleration will give the force required to accelerate them in poundals, while by dividing by g the result is in engineering units of pounds. It is to be noted that the acceleration of the piston is felt in all of the water in the system, and is inversely proportional to the areas, as is the case with the velocities. From these the following formulæ, may be derived, remembering that the force is exerted over the areas, A, A_1, A_2, etc.:

$$wAh_4 = \frac{AXwa}{g}; \quad \ldots \ldots \ldots \quad (17)$$

$$wA_2h_4 = \frac{A_2L_2wa_2}{g} = \frac{A_2L_2wAa}{gA_2}; \quad \ldots \ldots \quad (18)$$

$$wA_1h_4 = \frac{A_1L_1wa_1}{g} = \frac{A_1L_1wAa}{gA_1}. \quad \ldots \ldots \quad (19)$$

These are typical terms, and if there are a series of changes in the lines these reduce to

$$h_4 = \frac{a}{g}\left(X + \Sigma\frac{AL}{A_2} + \Sigma\frac{AL}{A_1}\right). \quad \ldots \ldots \quad (20)$$

The variation in pressure of the water on the bottom of the piston on the up stroke may now be written:

$$h_s = h_a - \left[h_2 + X \sin\phi + h_{sv} + \left(\Sigma\zeta_s\left(\frac{A}{A_1}\right)^2 + 1\right)\frac{S^2}{2g} \right.$$
$$\left. + \left(X + \Sigma\frac{AL}{A_2} + \Sigma\frac{AL}{A_1}\right)\frac{a}{g}\right]. \quad \ldots \quad (21)$$

where
$$X=r\left[1-\cos\theta+\frac{1}{2n}\sin^2\theta\right];$$

$$S=r\omega\left[\sin\theta+\frac{1}{2n}\sin 2\theta\right];$$

$$a=r\omega^2\left[\cos\theta+\frac{1}{n}\cos 2\theta\right].$$

The various terms of this formula which depend on experiment must now be discussed.

The first term to require explanation is the resistance h_{sv} of the suction valve. The discussion following is due to Hartmann and Knoke and is based on the work of Bach.

There are three types of valves, lift or beat valves, clack valves, and slide valves, as shown in Figs. 178, 179, and 180.

FIG. 178.—Lift Valve.

FIG. 179.—Clock Valve.

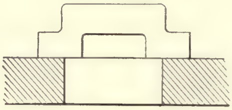


FIG. 180.—Slide Valve.

The valves serve the purpose of alternately connecting the pump cylinder to the suction or discharge pipe and with the exception of the last form of valve they are operated by the water of the pump. The slide valve is operated by some external means as an eccentric. At times, however, the clack or lift valve may be operated in part by some positive gear.

Now the resistance of the passage of water through valves as investigated by Bach is divided into two parts: (*a*) that due to the opening of the valve, and (*b*) that due to the passage of the water through the valves.

The valve has the velocity and acceleration of the water under it, the latter being $a\frac{A}{A_p}$, where A_p is the area of the passage beneath the valve. If the valve weight is W_v its inertia will be $\frac{W_v}{g}\frac{aA}{A_p}$. If the total spring pressure is taken

as F_s and the area of the upper side of the valve is A_v, the following equation exists between pressures above and below the valve:

$$p_l A_p = p_u A_v + W_v + F_s + \frac{W_v}{g}\frac{aA}{A_p}.$$

$$p_l = \frac{p_u A}{A_p} \quad \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v}{g}\frac{aA}{A_p^2}. \quad \ldots \quad (22)$$

$p_l - p_u =$ loss in pressure in opening valve.

$$= p_u\left(\frac{A_v - A_p}{A_p}\right) + \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v}{g}\frac{aA}{A_p^2}. \quad \ldots \quad (23)$$

On the upper side of the discharge valve the following relation holds:

$$p_u = \left[(h_a + H_d) + \frac{L_d}{g}\frac{A}{A_d}a\right]w, \quad \ldots \quad (23a)$$

where h_a represents the atmospheric pressure, H_d the static pressure in the pipe line, and $\frac{L_d}{g}\frac{A}{A_d}a$ the force required to accelerate the water in the discharge pipe.

This value may be determined in any given case before finding the quantity h_{dv} given below.

This may be expressed in head as

$$h_{dv} = \frac{p_l - p_u}{w} = \frac{1}{w}\left[p_u\left(\frac{A_v - A_p}{A_p}\right) + \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v}{g}\frac{A}{A_p^2}\right]. \quad (24)$$

For the suction valve this equation becomes:

$$p_l A_p = p_u A_v + W_v + F_s + \frac{W_v}{g}a\frac{A}{A_p}.$$

$$h_{sv} = \frac{1}{w}\left[p_u\left(\frac{A_v - A_p}{A_p}\right) + \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v}{g}a\frac{A}{A_p^2}\right], \quad \ldots \quad (25)$$

an expression exactly similar in form. In this case, however, it is better to find p_l originally by the formula:

$$p_l = \left[(h_a - H_s) - \frac{L}{g}\frac{A}{A_s}a\right]w. \quad \ldots \quad (26)$$

(H_s=head to suction valve and L=length to this point) and then p_u by the formula:

$$A_p p_l = A_v p_u + W_v + F_s + \frac{W_v}{g}\frac{A}{A_p}a. \quad . \quad . \quad . \quad (27)$$

Then

$$h_{sv} = \frac{p_l - p_u}{w} = \frac{1}{w}\left[p_l \frac{A_v - A_p}{A_v} + \frac{W_v}{A_v} + \frac{F_s}{A_v} + \frac{W_v}{g}\frac{Aa}{A_v A_p}\right]. \quad (28)$$

Before numerical values can be found for these expressions the value of F_s must be determined. The Eqs. (24) and (28) show the dependence of the pressure required to open the valve on the pressure above the valve, the weight of the valve, the spring pressure, and the acceleration which depends on the speed of the pump in revolutions per minute. The resistance or pressure increases with all of these. The effect of area in

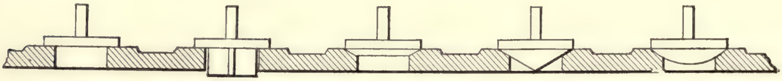


FIG. 181.—Valve Forms.

decreasing the value of this resistance shows what an important part the valve area plays in the design of pumps.

The resistance of valves to the flow of water beneath them has been found to be given by the expression

$$h_v = \zeta \frac{c_1^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

where c_1 =velocity of water in passage leading to valves;
h_v =the loss in feet head;
ζ =a coefficient, the value of which is given by formulæ below, depends on the form of valve and has been investigated for the forms shown in Fig. 181 in which Form I is a plain disc valve with no guides, Form II is a disc with guide vanes attached to the lower face, Form III the conical valve with no lower guides, Form IV a complete conical valve, and Form V a spherical valve.

$$\zeta=\alpha+\beta\left(\frac{d_1}{h}\right)^2, \text{ for valves of Forms I and IV.} \quad . \quad . \quad (30)$$

$$\zeta=\alpha+\beta\left[\frac{d_1^2}{(\pi d_1-is)h}\right]^2, \text{ for valves of Form II.} \quad . \quad . \quad (31)$$

$$\zeta=\alpha+\beta\frac{d_1}{h}+\gamma\left(\frac{d_1}{h}\right)^2, \text{ for valves of Forms III and V.} \quad (32)$$

Bach gives the various values for the constants and proportions for the parts which reduce to the forms below:

For Form I:

$$h=\frac{d_1}{10} \text{ to } \frac{d_1}{4};$$

$$\alpha=0.55+4\frac{b_1-0.1d_1}{d_1};$$

$$\beta=0.15 \text{ to } 0.16.$$

h = rise of valve in feet;
d_1 = diameter of passage leading to valve;
b_1 = breadth of seat in feet $=\frac{d_1}{10}$ to $\frac{d_1}{4}$;
i = number of ribs;
S = thickness of ribs at end.

For Form II:

$$h=\frac{d_1}{8} \text{ to } \frac{d_1}{4};$$

$$b_1=\frac{d_1}{10} \text{ to } \frac{d_1}{4};$$

$\alpha=\alpha$ for Form I multiplied by 0.8 to 1.6;
$\beta=1.70$ to 1.75.

For Form III:

$$h=\frac{d_1}{10} \text{ to } \frac{d_1}{4}, \quad b_1=0.1d_1;$$

$$\alpha=2.6;$$

$$\beta=-0.8;$$

$$\gamma=0.14.$$

For Form IV:

$$h=\frac{d_1}{8} \text{ to } \frac{d_1}{4};$$
$$\alpha=0.6;$$
$$\beta=0.15;$$

For Form V:

$$h=\frac{d_1}{10} \text{ to } \frac{d_1}{4};$$
$$\alpha=2.7;$$
$$\beta=-0.8;$$
$$\gamma=0.14.$$

The values of these coefficients were determined when the clear space between the valves was 1.8, the area of the passages between the valves, or

$$\frac{\pi}{4}(d_2^2-d^2)=1.8\frac{\pi}{4}d_1^2;$$

d_2 = diameter of space to point midway between valves,
d = diameter of valve, and
d_1 = diameter of passage.

With these coefficients the loss through the valve may be determined if the value c_1 is known. This will vary with the speed of the plunger.

The discharge area beneath a valve changes as the valve lifts, and it may change so that the velocity under the valve, designated by c, may remain constant.

$$AS=A_pc_1=\gamma c\pi dh, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

where A = area of piston or plunger;
S = speed of plunger at any instant;
A_p = area of passage beneath valve;
c_1 = velocity of water in passage beneath valve;
γ = coefficient of discharge;
c = velocity of discharge radially;
d = outside diameter of valve;
h = lift of valve.

In this case, however, the movement of the valve itself will cause a change in the velocity c or in the height h, because a quantity of water $\frac{\pi d^2}{4}v$, where v is the velocity of the valve, is held beneath the valve during the lifting of it or is discharged on the dropping of the valve. This then gives

$$A S = A_p c_1 = \gamma c \pi d h \pm A_v v, \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

calling $\pi d = l$,

$$A S = A_p c_1 = \gamma c l h \pm A_v v \quad . \quad . \quad . \quad . \quad . \quad . \quad (35)$$

(+ during rise of valve; − during fall.)

This means that the motion of the water in the passage below the valve relative to the valve is $c_1 \pm v$, and therefore

$$h = \zeta \frac{(c_1 \pm v)^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (36)$$

Hartmann and Knoke investigated this loss during the stroke of a pump, finding the values of h at different parts of a stroke experimentally by an indicator whose pencil motion was attached to the valve. From this they computed ζ by Eq. (30) at various points. They then computed the velocity of the water c and from the curve of movement of the valve determined v. The computation of h showed that the head over a large part of the stroke was practically constant, the variations being at the end. The curve found was similar to Fig. 182. This showed that the loss could be computed as if the water were moving at the speed determined by the crank position of 90° with a zero velocity of the valve or at a point where the valve has reached the top of its stroke.

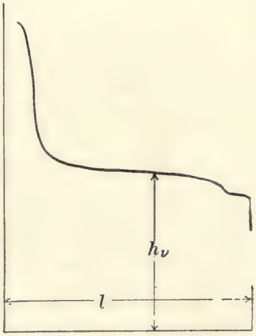


FIG. 182.—Valve Friction.

Here

$$c_1 = \frac{A S_{90}}{A_p} = \frac{A \omega r}{A_p}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

and

$$h_v = \left(\frac{A\omega r}{A_p}\right)^2 \frac{\zeta}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

Now

$$h_v = \frac{Fs + W_v}{A_p w} = \zeta \frac{c_1^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

where F_s is the spring tension, W_v is the weight of the valve, and h_v is the resistance from the valve expressed in feet of water. From this

$$c_1 = \frac{1}{\sqrt{\zeta}} \sqrt{2g \frac{W_v + F_s}{A_p w}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

Now, $\quad c_1 A_p = \gamma c l h,$

where γ is the coefficient of discharge beneath the valve and c is the radial velocity through the valve opening of lift h.

$$\therefore \qquad c = \frac{c_1 A_p}{\gamma l h} \quad \text{or} \quad \frac{1}{\sqrt{\zeta}} \sqrt{2g \frac{W_v + F_s}{A_v w}} \frac{A_p}{\gamma l h}, \quad . \quad . \quad (41)$$

or using the area A_v of the valve and a new coefficient by analogy this becomes

$$c = \frac{1}{\sqrt{\zeta_1}} \sqrt{2g \frac{W_v + F_s}{A_v w}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

It is to be noted that this is expressed in terms of feet head of water exerted over the complete valve area A_v.

The equation then given for h is

$$h = \frac{c_1 A_v}{\gamma l c} = \frac{c_1 A_v}{\frac{\gamma l}{\sqrt{\zeta_1}} \sqrt{2g \frac{W_v + F_s}{A_v w}}},$$

or

$$h = \frac{c_1 A_v}{\mu l \sqrt{2gb}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

where b expresses the feet of water equivalent to the pressure on the whole area of the valve and $\mu = \frac{\gamma}{\sqrt{\zeta_1}}$.

Since
$$\dot{h}=\frac{c_1A_v}{\gamma lc};$$

$$\gamma lc=\mu l\sqrt{2gb},$$

or

$$\gamma c=\mu\sqrt{2gb}. \qquad (44)$$

The formulæ just used were for definite velocities, but actual velocities are variable. This will change certain of the forms, as observed before in the discussion of the resistance to the passage of the water.

In the first place the movement of the valve and its change in velocity introduces a change in the term b, which has been used to denote

$$\frac{W_v+F_s}{A_vw}.$$

On account of friction of the valve and its inertia

$$P=W_v+F_s\pm R+\frac{W_va_v}{g}, \qquad (45)$$

where R = resistance due to friction and $\frac{W_va_v}{g}$ = force required to accelerate the valve.

By properly designing the valve and its connections the term R may be made small and the term $\frac{W_va_v}{g}$ is small in slow-running pumps on account of the acceleration being small, and in fast-running pumps F_s is large, so that $\frac{W_va_v}{g}$ is small in comparison with W_v+F_s. Hence, although not strictly correct, the resistance to valve motion may be considered as W_v+F_s or when measured in feet of water on area of valve, $\frac{W_v+F_s}{A_vw}$.

Now
$$F_s=c(y_0+h),$$

where c = the force required to compress the spring one foot provided the elasticity is constant;

y_0 = the initial compression of the spring in feet;

h = the lift of the valve in feet.

This quantity depends on the value of h, and except in cases where y_0 is large in comparison with the maximum lift the variation in F_s and consequently b must be considered.

Since the valve is in motion the equation for discharge through the valve is

$$\gamma clh = A_p c_1 - A_v v.$$

However,

$$A_p c_1 = AS = Ar\omega\left[\sin\theta + \frac{1}{2n}\sin 2\theta\right].$$

Hence

$$h = \frac{Ar\omega\left[\sin\theta + \frac{1}{2n}\sin 2\theta\right] - A_v v}{\gamma lc}. \quad . \quad . \quad (46)$$

but $\gamma c = \mu\sqrt{2gb}$ and velocity of valve

$$v = \frac{dh}{dt} = \frac{d}{dt}\left(\frac{AS - A_v v}{\mu l\sqrt{2gb}}\right)$$

$$= \frac{1}{\mu l\sqrt{2gb}}\left(A\frac{ds}{dt} - A_v\frac{dv}{dt}\right).$$

Now the acceleration of the valve $\frac{dv}{dt}$ is a small quantity, since the lift and velocity of the valve are small, and consequently it may be neglected, giving

$$v = \frac{A}{\mu l\sqrt{2gb}}\frac{ds}{dt}.$$

Hence

$$h = \frac{Ar\omega\left[\sin\theta + \frac{1}{2n}\sin 2\theta\right] - \frac{A_v A r\omega^2}{\mu l\sqrt{2gb}}\left[\cos\theta + \frac{1}{n}\cos 2\theta\right]}{\mu l\sqrt{2gb}}$$

$$= \frac{Ar\omega}{\mu l\sqrt{2gb}}\left\{\left(\sin\theta + \frac{1}{2n}\sin 2\theta\right) - \frac{A_v\omega}{\mu l\sqrt{2gb}}\left(\cos\theta + \frac{1}{n}\cos 2\theta\right)\right\} \quad (47)$$

From this equation it is seen that the valve does not come to its seat when the pump is on dead center, nor does it leave its seat at that time. It is seen, however, from the curves of discharge from a pump that the rate of discharge remains

almost constant near the crank position of $\theta=90°$, and it may be assumed that this value gives the maximum value of h. This gives three important values to find, namely, h_0, $h_{max}=h_{90°}$, and the angle θ, at which $h=0$.

$$h_{0°}=-\frac{Ar\omega}{\mu l\sqrt{2gb}}\left\{\frac{A_v\omega}{\mu l\sqrt{2gb}}\left(1+\frac{1}{n}\right)\right\}=-\frac{AA_vr\omega^2}{\mu^2l^2 2gb}\left(1+\frac{1}{n}\right) \quad (48)$$

$$h_{180°}=\frac{AA_vr\omega^2}{\mu^2l^2 2gb}\left(1-\frac{1}{n}\right). \quad (49)$$

$$h_{max}=h_{90°}=\frac{Ar\omega}{\mu l\sqrt{2gb}}\left\{1+\frac{1}{n}\frac{A_v\omega}{\mu l\sqrt{2gb}}\right\}. \quad (50)$$

$$=h_{270°}=-\frac{Ar\omega}{\mu l\sqrt{2gb}}\left\{1-\frac{1}{n}\frac{A_v\omega}{\mu l\sqrt{2gb}}\right\}. \quad (51)$$

The values for 0° and 180° and for 90° and 270° are different because of the effect of the angularity of the connecting rod, which causes a difference in the action of the two ends. The values 180° and 90° refer to the stroke from the head to the crank, while the others refer to the return stroke. The mean values of these pairs

$$h_0=\frac{AA_vr\omega^2}{\mu^2l^2 2gb} \quad (52)$$

$$h_m=\frac{Ar\omega}{\mu l\sqrt{2gb}} \quad (53)$$

are the values which would be obtained were the value of n infinite and there were no effect of angularity.

Suppose that the valve reaches its seat at a value $\theta=180°+\delta$, then

$$h=h_\delta=0=\frac{Ar\omega}{\mu l\sqrt{2gb}}\left\{\left[\sin(180°+\delta)+\frac{1}{2n}\sin(360°+2\delta)\right]\right.$$
$$\left.-\frac{A_v\omega}{\mu l\sqrt{2gb}}\left[\cos(180°+\delta)+\frac{1}{n}\cos(360°+2\delta)\right]\right\}.$$

$$\frac{A_v\omega}{\mu l\sqrt{2gb}}=\frac{-\sin\delta+\frac{1}{2n}\sin 2\delta}{-\cos\delta+\frac{1}{n}\cos 2\delta}.$$

For a small angle δ:

$$\sin 2\delta = 2 \sin \delta \cos \delta = 2 \sin \delta.$$
$$\cos 2\delta = \cos^2 \delta - \sin^2 \delta = \cos^2 \delta = \cos \delta$$

$$\frac{A_v\omega}{\mu l\sqrt{2gb}} = \frac{\sin \delta - \frac{1}{n} \sin \delta}{\cos \delta - \frac{1}{n} \cos \delta} = \tan \delta \text{ (approx.)}. \quad . \quad (54)$$

At the point where $h=0$ the velocity of the valve is not given by the formula for velocity used in reducing the value of h, because at this point the acceleration $\frac{dv}{dt}$ of the valve is not negligible, and hence the following formula is used:

$$\gamma hcl = AS - A_v v$$
$$0 = AS - A_v v,$$
$$v_0 = \frac{AS}{A_v} = \frac{Ar\omega}{A_v}\left[\sin (180° + \delta) + \frac{1}{2n} \sin (360° + 2\delta)\right]$$
$$= \frac{Ar\omega}{A_v}\left(-\sin \delta + \frac{1}{2n} \sin 2\delta\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (55)$$

This velocity is theoretic and is dissipated by a water cushion. Hartmann and Knoke experimented on valve seating by attaching an indicator pencil to a valve stem. The line drawn when the drum was rotated showed that although there was impact and shock when the valve seated in air, this did not occur when water was present with a weight-driven or a spring-driven valve. This would then permit of considerable velocity if the water has no chance to pass back into the cylinder. If, however, the value of δ is largely due to a large value of ω, the closing of the valve is brought about by the removal of water through the channel which formerly brought water to the valve and this will cause hammering of the valves, although the water sucked in from the discharge or forced back through the suction will dampen this to a certain extent.

The calculation for the size of valves and the pressures of the springs depends on the coefficient $\mu = \frac{\gamma}{\sqrt{\zeta_1}}$, where γ was the coefficient of discharge from the edge of the valve and

$$\zeta_1 = \alpha + \beta\left(\frac{d_1}{h}\right)^2,$$

where α and β are Bach coefficients. The quantities, especially γ, depend on the velocity c, which has been considered as constant and is not quite so.

Experiments have been resorted to, to get the value of μ. The values of h_0 and $h_{\max}$ are given by the equations:

$$h_0 = \frac{AA_v r\omega^2}{(\mu l\sqrt{2gb})^2} = \frac{Q\pi\omega A_v}{(\mu l\sqrt{2gb})^2}. \quad . \quad . \quad . \quad . \quad (56)$$

Since

$$A2r\frac{N}{60} = Q = A2r\frac{\omega}{2\pi},$$

or

$$Q\pi = Ar\omega.$$

$$h_m = \frac{Ar\omega}{\mu l\sqrt{2gb}} = \frac{Q\pi}{\mu\sqrt{2gbl}}. \quad . \quad . \quad . \quad . \quad . \quad (57)$$

At the dead point there is no discharge from the pump cylinder and the discharge from the valve is due to the spring pressure at this point. Here v varies as $\sqrt{2gb}$.

$$\gamma clh_0 = 0 - A_v v,$$

or

$$\frac{\gamma c}{v} = -\frac{A_v}{lh_0} = -\frac{\frac{\pi d^2}{4}}{\pi dh_0} = -\frac{d}{4h_0},$$

$$\mu\sqrt{2gb} = -\frac{d}{4h_0}v.$$

From this it is seen that μ varies as $\frac{d}{4h_0}$ for different valves if the head b is to produce the velocity v.

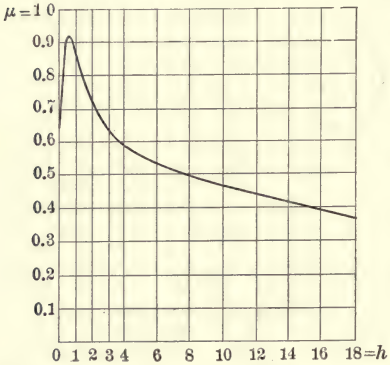


FIG. 183.—Variation of μ with h, according to Hartman and Knoke.

In these equations for a given valve and spring, and pump it would be only necessary to operate the pump at different speeds, measuring h_0 or h_{max}, and from them computing μ as all other quantities could be measured Hartmann and Knoke have experimented, finding the values of μ for different values of h. This gives a curve of the form shown in Fig. 183.

TABLE I

VALUES OF μ

h		$\frac{d}{4h}$	μ	h		$\frac{d}{4h}$	μ
mm.	ins.			mm.	ins.		
0.0	0.000	....	0.650	6.0	0.236	2.50	0.532
.1	.004	150.00	.710	6.5	.256	2.31	.523
.2	.008	75.00	.780	7.0	.276	2.14	.515
.3	.012	50.00	.845	7.5	.295	2.00	.507
.4	.016	37.50	.890	8.0	.315	1.87	.500
.5	.020	30.00	.911	8.5	.335	1.76	.493
.6	.024	25.00	.913	9.0	.354	1.67	.485
.8	.031	18.75	.902	9.5	.374	1.58	.477
1.0	.039	15.00	.870	10.0	.394	1.50	.472
1.5	.059	10.00	.788	11.0	.433	1.36	.459
2.0	.079	7.50	.732	12.0	.472	1.25	.445
2.5	.098	6.00	.690	13.0	.512	1.15	.431
3.0	.118	5.00	.650	14.0	.551	1.07	.420
3.5	.138	4.28	.622	15.0	.591	1.00	.407
4.0	.157	3.75	.599	16.0	.630	.94	.395
4.5	.177	3.33	.578	17.0	.669	.88	.381
5.0	.197	3.00	.560	18.0	.709	.33	.370
5.5	.217	2.73	.545				

(From Hartmann-Knoke, "Die Pumpen.")

The experimenters computed these values for each value of h_{max} as if it were an h_0. Their reason for this being that the value of $\frac{d}{4h_0}$ is the quantity which would be used in valve design of various sizes of valves, and the quantity μ of the expression for velocity depends on this quantity $\frac{d}{4h_0}$. The desire in any design is to have as large a value for μ as possible, and it is seen from Fig. 183 and Table I that this occurs when the value of $\frac{d}{4h}$ varies from 15 to 37.5; μ varying from 0.87 to 0.913.

If $\frac{d}{4h_0}$ be taken as 50, so as to reduce h_0,

$$h_0=\frac{1}{200}d, \text{ and } \frac{\gamma c}{v}=50. \quad \text{From the table}$$

$$\mu=.845.$$

Hartmann and Knoke suggest that h_0 be made $h_0=\frac{1}{250}d$. This gives the following:

$$h_0=\frac{1}{250}d;$$

$$\frac{\gamma c}{v}=62.5;$$

$$\mu=.80.$$

This value of μ, however, is smaller than the value first selected, and the valves would have to be larger so that the other proportion will be used.

Now, as given in equation (56)

$$h_0=\frac{Q\pi\omega A_v}{(\mu\sqrt{2gb_0l})^2}.$$

Substituting the above values for μ and simplifying the following results:

$$h_0=\frac{Q\pi\frac{N\pi}{30}\frac{\pi d^2}{4}}{(.85\sqrt{2gb_0\pi d})^2}=\frac{\pi QN}{120\times.72\times2gb_0}=\frac{1}{200}d.$$

$$b_0d=0.1124QN. \quad \ldots\ldots\ldots \quad (57a)$$

The $\frac{d}{4}$ in the expression $\frac{d}{4h}$ came from the reduction of $\frac{A_v}{l}=\frac{d}{4}$, hence

$$h_0=\frac{1}{50}\frac{d}{4}=\frac{1}{50}\frac{A_v}{l};$$

but

$$h_0=\frac{Q\pi\omega A_v}{(\mu\sqrt{2gb_0}l)^2};$$

and

$$\frac{1}{50}\frac{A_v}{l}=\frac{Q\pi\omega A_v}{\mu^2 2gb_0 l^2},$$

$$lb_0=\frac{\pi^2 50}{60\times.72\times g}QN=0.3556QN. \quad . \quad (57b)$$

Eq. (56) mav be reduced to a simpler form if the quantity ω is expressed in terms of N, giving

$$\frac{h_0 l^2 b_0}{A_v}=\frac{Q\pi^2 N}{.72\times 30\times 2g}=0.0071QN. \quad . \quad . \quad . \quad (57c)$$

These three formulæ may be used to determine the dimensions of the valves, when b_0, d, h_0, l, or A is assumed and the other quantities may be found by their use. The expressions appear as if all of the water is taken through one valve, and if n valves are used the quantity Q is replaced by $\frac{Q}{n}$. If the diameter d is made larger, the quantity h_0 becomes smaller, since $\frac{d_0}{4h}$ is taken equal to 50. To cut down the loss through the valves the quantity b_0 is made as small as possible. The value of b_0 fixes the velocity of discharge under the valve at the dead point, since

$$\gamma c=\mu\sqrt{2gb_0},$$

and this quantity b_0 should be as small as it can be to give c a proper value. The value of 3 feet for b_0 may be used.

After b_0 is assumed it is necessary to determine b_{max},

$$h_{max}=\frac{Q\pi}{\mu\sqrt{2gb_{max}}l}=\frac{Q}{\mu\sqrt{2gb_{max}}d}. \quad . \quad . \quad . \quad (58)$$

To find the value of b_{max}, assume the value of h_{max} (larger than h_0), from this find the value of $\frac{d}{4h}$ and the proper value of μ and then the value of b_{max}. These then give the values of the force on the valve at opening and when full open, and from them the constants of the spring may be found.

$$b_0 = \frac{G+(y_0+h_0)C}{A_v w}; \quad . \quad . \quad . \quad . \quad . \quad (59)$$

$$b_m = \frac{G+(y_0+h_{max})C}{A_v w}. \quad . \quad . \quad . \quad . \quad . \quad (59a)$$

G is the weight of the valve, y_0 the initial amount of compression, and C the force required to compress the spring a unit amount. These two equations will give y_0 and C from which the spring may be designed.

$$C = A_v w \frac{b_{max} - b_0}{h_{max} - h_0}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (60)$$

To apply these formulæ, it will be required to find the valves for the pump for which the data have been given. The pump was 20×30 inches, making 60 R.P.M.

The American method of designing pumps is to use a large number of small valves in place of a small number of large valves.

The reason for this is the fact that since the area of a valve varies as the square of its diameter, and its discharge area varies with the first power only, smaller valves have a larger discharging area for the same area of valve deck. To utilize the central part of large valves, they are made multiported, as shown in Fig. 184. It is this type of valve which will be compared with the small American valves:

$$Q = 2rA\frac{N}{60} = \frac{30 \times (314.16 - 3.14)}{1728} \times \frac{60}{60} = 5.44 \text{ cu. ft.}$$

(The piston of the water end is assumed to be carried by a 2-inch piston rod and a 2-inch tail rod.)

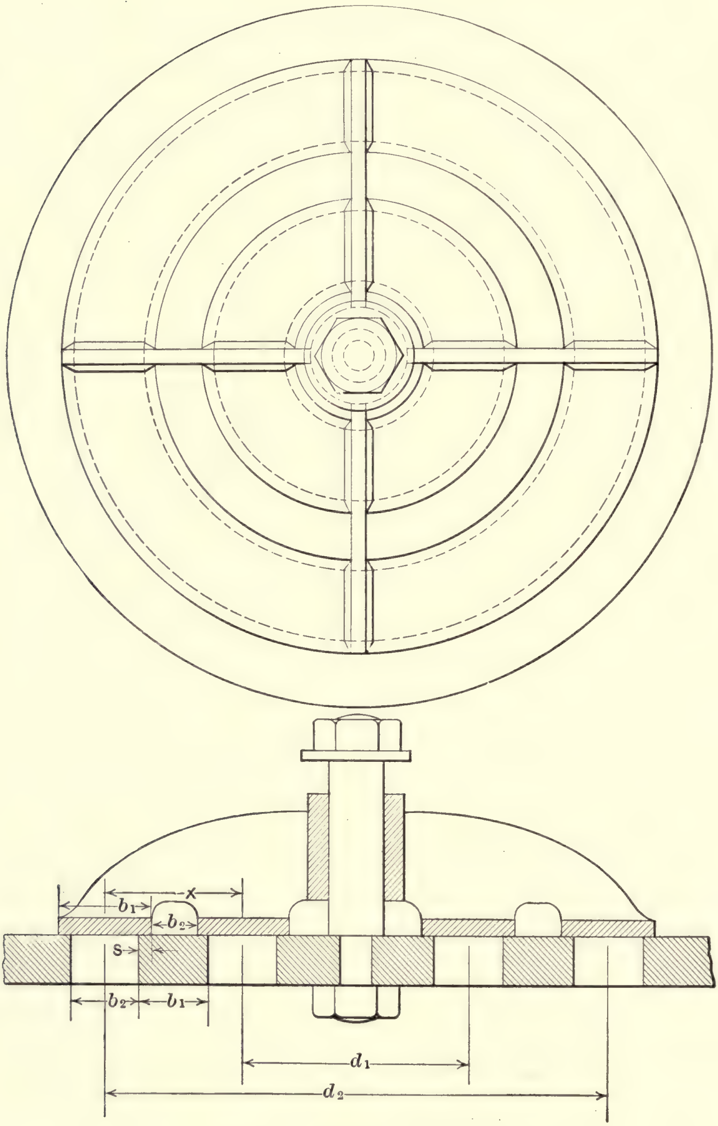


FIG. 184.—Multiported Valve.

American practice is to use 4-inch valve, and this diameter will be assumed.

$$b_0=\frac{0.1124\frac{Q}{n}N}{d};$$

assuming

$$b_0=1 \text{ ft.}$$

$$n=\frac{0.1124\times 5.44\times 60}{1\times .33}=112 \text{ valves};$$

$$h_0=\frac{1}{200}d=\frac{.33}{200}=0.00167 \text{ ft.}$$

For the multiported valve several formulæ will be derived.

$$A_v=\pi d b_1$$

for each;

$$\Sigma A_v=\Sigma\pi d b_1=\pi b_1\Sigma d;$$

$$l=2\pi d; \quad \ldots \quad (61)$$

$$\Sigma l=2\pi\Sigma d. \quad \ldots \quad (62)$$

Hence

$$\frac{A_v}{l}=\frac{\Sigma A}{\Sigma l}=\frac{\pi b_1\Sigma d}{2\pi\Sigma d}=\frac{b_1}{2}; \quad \ldots \quad (63)$$

$$h_0=\frac{1}{50}\frac{A_v}{l}=\frac{b_1}{100}.$$

From Fig. 184

$$b_2+b_1=x;$$

$$b_1-b_2=2s.$$

Now $b_0l=.3556QN$.

Assuming, as before, $b_0=1$ ft,

$$l=\frac{.3556\times 5.44\times 60}{1}=117;$$

$$l=2\pi\Sigma d;$$

$$\Sigma d=\frac{117}{2\pi}=18.5.$$

From the figure

$$\frac{\pi}{4}(d_1-x)^2+\pi d_1x+\pi d_2x+\ .\ .\ .\ =\frac{\pi}{4}[d_1+(2n-1)x]^2;$$

or

$$\pi x\Sigma d=\pi nd_1x-\pi nx^2+\pi n^2x^2$$

$$x=\frac{\Sigma d-nd_1}{n^2-n}=\frac{\Sigma d-nd_1}{n(n-1)}.\qquad .\ .\ .\ .\ .\ (64)$$

Assuming $d_1=6$ in. and $n=10$,

$$x=\frac{18.5-10\times\frac{1}{2}}{100-10}=.15\text{ ft.}$$

Let $$b_1=.08\text{ ft.};$$

$$b_2=.07\text{ ft.};$$

$$d_{out}=d_1+(2n-1)x=\tfrac{1}{2}+19\times.15=3.35;$$

$$h_0=\frac{1}{100}b_1=\frac{0.08}{100}=0.0008.$$

Comparing these two results, assuming 50 per cent as the excess area required for valve deck, it is seen that the 112 valves require 14.64 square feet of area while the multiported valve requires 13.2 square feet.

This calculation shows the necessity of a large number of valves when a high-speed pump is used. If the speed of the pump is decreased the number of valves will be diminished in proportion to the square of the speed, as the number varies as the product QN, in which Q depends on N. Thus for a 20-revolution pump the number of valves would be decreased to $\frac{1}{9}\times112=12$, and the diameter of the multiported valve would be changed to $\frac{1}{3}\times3.35$, or 1.12 feet. The area required would be 1.63 square feet and .98 square foot, respectively, for the small valves and the multiported valve. By changing b_0 the number and size could be diminished. This, with a

2-foot pressure for b_0 in the original calculation, results in the following:

$$n = 56.$$
$$d_0 = 1.7 \text{ approx.}$$
$$\text{Area valve deck} = 10.00 \text{ sq. ft.}$$
$$\text{Area multiported valve} = 9.4 \text{ sq. ft.}$$

To find the spring pressure at the position h_0 use the following formula:

$$wb_0A_v = W_v + F_0.$$

For small valve

$$62.5 \times 1 \times \frac{4\pi}{144} = W_v + F_0;$$

assuming $W_v = \frac{1}{2} lb$,

$$5.45 = \tfrac{1}{2} + F_0;$$
$$\therefore\ F_0 = 4.95.$$

For large valves

$$62.5 \times 1 \times \pi \times .08 \times 18.5 = W_v + F_0;$$
$$287. = W_v + F_0.$$

Let

$$W_v = 90 \text{ lb.};$$
$$\therefore\quad F_0 = 197;$$

Now

$$h_m = \frac{Q\pi}{\mu l \sqrt{2gb_m}}.$$

Since h_{max} will be greater than h_0, consider it as .025 for the first approximation for small valves.

$$\frac{d}{4h} = \frac{.33}{.10} = 3.3;$$

$$\mu = .63;$$

$$b_m = \left(\frac{Q}{n\mu\sqrt{2gdh_m}}\right)^2 = \left(\frac{5.44}{112 \times .63 \times 8.02 \times .33 \times .025}\right)^2 = 1.16 \text{ ft.}$$

For the large valve let

$$h_m = .02$$

so as to give less distance for this large valve to seat,

$$\frac{b_0}{2h_m}=\frac{.08}{2\times .02}=2.0.$$

Hence

$$\mu=.73;$$

$$b_{\max}=\left(\frac{Q\pi}{\mu\sqrt{2g}\,\Sigma l h_{\max}}\right)^2$$

$$=\left(\frac{5.44}{.73\times 8.02\times 2\times 18.5\times .02}\right)^2=1.58 \text{ ft.}$$

Since

$$\Sigma l=2\pi\times 18.5;$$

$$W_v+F_0=62.5\times 1.58\times\pi\times .08\times 18.5=453.$$

In the case of the small valve, the spring tension which was $5.45-.5$ at .0017 movement will be $7.15-.5$ or 6.65 at .025 movement, or $\dfrac{6.05-4.95}{(.025-.0017)12}=4$ lb. per inch of compression (approx.).

For the large valve this becomes

$$\frac{(453-90)-197}{(.02-.0008)12}=\frac{166}{.0192\times 12}=720 \text{ lb. per inch, approx.}$$

In this case there would be several springs to hold such a large valve to its seat, say ten. This would give 72 lb. per inch for the modulus of these springs. The large valve as designed would be very cumbersome and would undoubtedly be divided among a number of smaller valves, changing the results considerably. It is suggested that the student work out the data, using six valves of the same style in place of one large valve.

The method of designing the springs will be considered in the next chapter.

To compare the results for the 4-inch valves with another method used to determine the number of valves, consider the rule that the velocity through the area occupied by the valves should be between 100 and 125 feet per minute. Since the

water is delivered through the valves in one stroke or a half revolution, this gives

$$n=\frac{2Q\times 60}{100\times(\text{area})}=\frac{2\times 5.46\times 60}{100\times\frac{4\pi}{144}}=75.$$

This means about 200 to 250 feet per minute through the opening in the valve deck. If this is figured on the maximum velocity through the opening in place of the mean this number 75 will have to be increased by the ratio of $\pi:2$ or $n=\frac{\pi}{2}\times 75=118$. This result is not very different from the previous result.

Others design the total valve area to be a certain percentage of the piston area. Mr. C. A. Hague recommends using 50 per cent of the piston area when the piston speed is 100 feet per minute and 150 per cent when the piston speed is 300 feet per minute. For intermediate piston speeds an intermediate value of the percentage is used. Mr. I. H. Reynolds uses a very simple rule: 1 square foot of valve area is required on suction and on discharge for each million gallons capacity. In this way the piston speed need not be considered, as it is the quantity of water in a given time which fixes the area of the valves.

Applying these two methods, the following results:

Piston speed $2LN=2\times\frac{30}{12}\times 60=300.$

Hague's method:

Area piston $=2.16$ sq. ft.

Valve area $=1.50\times 2.16=3.24$ sq. ft.

Number of valves $=\frac{3.24}{0.06}=54$ valves.

Reynolds' method:

Capacity pump $\frac{5.44\times 60\times 60\times 24\times 1728}{231}=3{,}515{,}000$ gallons per 24 hours.

Area $=3.52$ sq. ft.

Number of valves $\frac{3.52}{0.06}=59$ valves.

The resistance of the 4-inch valves during the operation of the pump is

$$h_v = \zeta \frac{C_p^2}{2g} = \frac{\zeta}{2g}\left(\frac{A\omega r}{A_p}\right)^2.$$

For disc valves

$$\zeta = \alpha + \beta\left(\frac{d_p}{h}\right)^2$$

$$= .55 + 4\,\frac{0.029 - 0.029}{0.29} + .15\left(\frac{0.29}{.025}\right)^2$$

$$= .55 + 0 + 18.2 = 18.7$$

$$h_v = \frac{18.7}{64.32}\left(\frac{2.16}{7.39}\,2\pi\,\frac{15}{12}\right)^2 = 1.36 \text{ ft.}$$

The resistance to water passage through both the suction and discharge valves is the same. The resistance, however, at each end of the stroke is different for these, as at that time inertia plays an important part, and this effect depends on the length of water column, which is greater on the discharge side. The equations

$$h_{dv_0} = \frac{1}{w}\left[p_u\frac{A_v - A_p}{A_p} + \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v a}{g}\,\frac{A}{A_p^2}\right]$$

and

$$h_{sv} = \frac{1}{w}\left[p_u\frac{A_v - A_p}{A_p} + \frac{W_v}{A_p} + \frac{F_s}{A_p} + \frac{W_v a}{g}\,\frac{A}{A_p^2}\right];$$

give the loss of pressure at opening of the valves, and to get numerical values for these the pumps with the 4-inch valves will be considered, assuming that the suction lift is equivalent to 10 feet, including friction; that the discharge head is equivalent to 232 feet with friction; that there are 10 feet of 18-inch suction pipe and 200 feet of 18-inch discharge pipe, and that the 4-inch valves have $3\frac{1}{2}$-inch openings beneath them.

At the end of the stroke

$$a = \left(1 + \frac{1}{n}\right)\left(\frac{\pi N}{30}\right)^2 r;$$

$$= \frac{7}{6}\,\frac{\pi^2 60^2}{30^2}\,\frac{15}{12} = 58 \text{ ft. per sec. per sec.}$$

$$A_v = 112\frac{\pi 4^2}{4\times 144} = 9.74 \text{ sq ft.};$$

$$A_p = 112\frac{\pi}{4}.29^2 = 7.39 \text{ sq. ft.};$$

$$A = 2.16 \text{ sq. ft.};$$

$$A_d = \frac{\pi}{4}\frac{18^2}{144} = 1.77 \text{ sq. ft.}$$

From Eq. (23*a*)

$$p_{ud} = \left[34 + 232 + \frac{200}{32.16}\frac{2.16}{1.77}58\right]62.5$$

$$= 706\times 62.5 = 44{,}125 \text{ lb. per sq. ft.}$$

The effect of inertia in the pipe line is seen to be much greater than the static head. Of course this water probably does not start from rest each time and hence such a high value of p_{ud} may not occur. If this does not occur the water column must separate and pounding will be observed.

$$h_{dv_0} = \frac{1}{62.5}\left[44125\frac{9.74-7.39}{7.39} + \frac{112\times 0.5}{7.39} + \frac{4.95\times 112}{7.39}\right.$$

$$\left. + \frac{112\times 0.5}{32.16}58\cdot\frac{2.16}{7.39^2}\right] = 226 \text{ ft.}$$

This large resistance at opening of the valve is due to the fact that a large discharge pipe is used with no air chamber; the effect of such a chamber will be seen later.

$$p_{ls} = \left[34 - 10 - \frac{10}{32.16}\frac{2.16}{1.77}58\right]62.5 = 2\times 62.5 = 125.0 \text{ lb. per sq.ft.}$$

The result shows that the net pressure causing flow in the suction pipe is only 2 feet, and hence a slight increase in the speed of the pump would cause the suction column to break.

$$h_{sv_0} = \frac{1}{62.5}\left[125\frac{9.74-7.39}{9.74} + \frac{112\times 0.5}{9.74} + \frac{4.95\times 112}{9.74}\right.$$

$$\left. + \frac{112\times 0.5}{32.16}58\frac{2.16}{(9.74)(7.39)}\right] = 2 \text{ ft., approximately.}$$

These quantities h_{dv_0} and h_{sv_0} are the resistances at the time of opening the valves. Their values may be quite large, and in order to reduce them air chambers are used, as will be seen later. It is well to remember that p_s, or the pressure under the piston during the suction stroke, can never be less than zero. This means that the sum of the resistances and the lift on the suction side must be less than 34 feet. In many cases pumps are run so fast that this quantity is exceeded, when the water column will break and pounding will result.

In designing pumps with clack valves the same methods as used with the lift valves may be employed, remembering that moments of forces about the pivot must be considered and also that the perimeter of discharge is not the complete perimeter of the valve. The lift h may be taken as the lift at the outer end.

The next term of the pressure h_s Eq. (21) involves the hydraulic system.

For many years it has been customary to consider that the loss of head in a pipe line was given by Eq. (11).

$$h_p = f\frac{l}{d}\frac{v^2}{2g};$$

where h_p = loss of pressure expressed in feet;

f = a coefficient which varies with the kind of pipe, diameter, and velocity;

l = length of pipe in feet;

v = velocity in feet per second;

d = diameter of pipe in feet;

g = acceleration of gravity in feet per sec. per sec.

Since f varies with the velocity an attempt was made to eliminate from this term the velocity, and for this purpose the equation was put in the form Eq. (12):

$$h = kv^n l.$$

By plotting the results of many experimenters on logarithmic paper and by the methods of least squares Reynolds and others have found that n has a value of 1 for low velocities.

The value of n changes to about 1.75 when the velocity exceeds a certain amount. The point at which this change occurs is called the critical velocity. Reynolds showed, by injecting a colored liquid in a fine stream into the water in a glass tube, that so long as the velocity was less than the critical velocity the colored stream remained as a thread, showing that the water was traveling in stream lines; but as the critical velocity was reached the color became mixed with the other water, showing that there were eddies throughout the whole stream. Some have found this same effect by other means.

If the results of a great number of experimenters are examined, n will be found to have various values, varying from 1.7 to 2, rough pipe seeming to give the higher value. The value of k of the formula depends on d, and F. C. Lea in his "Hydraulics" shows that

$$k=K'd^{-1.25}.$$

This gives, then,

$$h=\frac{K'lv^{1.75}}{d^{1.25}} \quad \text{to} \quad \frac{K'lv^{2.0}}{d^{1.25}}.$$

There are thus two formulæ giving the loss in a pipe line, one in which the loss is expressed in terms of the square of the velocity and a factor which varies with the diameter and velocity, and another in which the loss is expressed in terms of powers of the velocity and diameter and a factor which is a constant for one kind of pipe.

For general purposes in cast-iron pipes the loss given by the two formulæ is by the approximate forms:

$$\left.\begin{aligned} h_s&=0.030\frac{l}{d}\frac{v^2}{2g} \\ h_s&=0.00055l\frac{v^2}{d^{1.25}} \end{aligned}\right\} \text{For ordinary cast-iron pipe, not clean.}$$

The value of f for clean pipe is given in tables in books such as Merriman's "Hydraulics" and Lea's "Hydraulics." The tabular value for f is about 0.03 for clean pipe when an average

value only is needed. For an exact value when the table is not at hand the following may be used:

$$f=\frac{0.026}{(vd)^{\frac{1}{4}}}.$$

These two expressions for loss may be put in the form $h_s=\zeta\frac{v^2}{2g}$ where ζ has the value $f\frac{l}{d}$ or $\frac{K'l}{d^{1.25}v^{.25}}$. In each case ζ has a value which varies with the velocity to a small degree, and as the other losses are expressed in terms of v^2, it is well to express this loss in the same manner.

The loss at entrance into a pipe line depends on the arrangement of the end. If a plain end flush with the wall of the fore bay is used the loss at entrance has been shown (Merriman's "Hydraulics," page 207) to be $0.49\frac{v_s^2}{2g}$, while if the pipe projects through the wall the coefficient becomes 0.93, and if a mouthpiece is used on the end of the pipe the coefficient is very small. In general it may be taken that the loss at entrance is given by the equation

$$h_e=\tfrac{1}{2}\frac{v^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (65)$$

The loss due to bends and obstructions, such as valves, is given in the form

$$h_b=kf\frac{lv^2}{d2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (66)$$

for bends, and

$$h=m\frac{v^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

for valves and cocks, where k is a multiplier of the f of straight pipe and has values depending on the ratio of the radius of the bend to the radius of the pipe and m on the amount the valve or cock is closed.

The values of k and m for the different cases are given in the forms of curves (Figs. 185, 186, 187). These have been

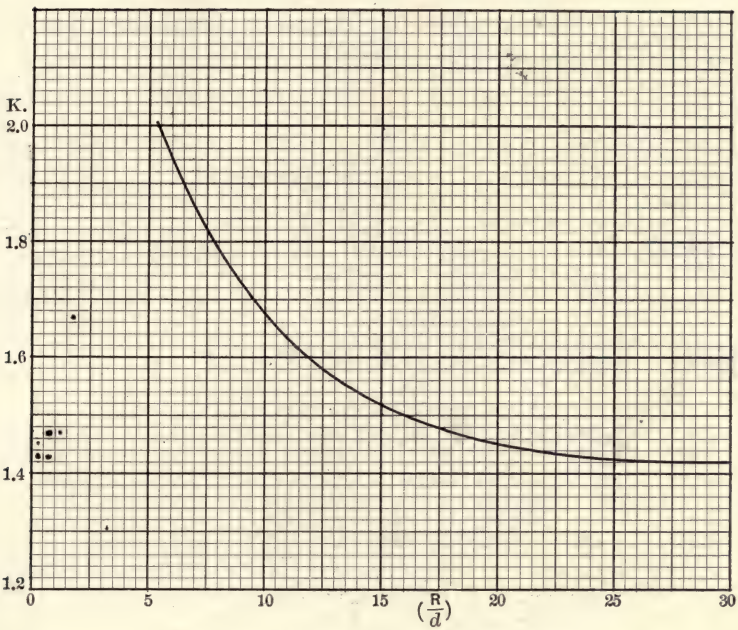


FIG. 185.—Values of K for Loss in Bends.

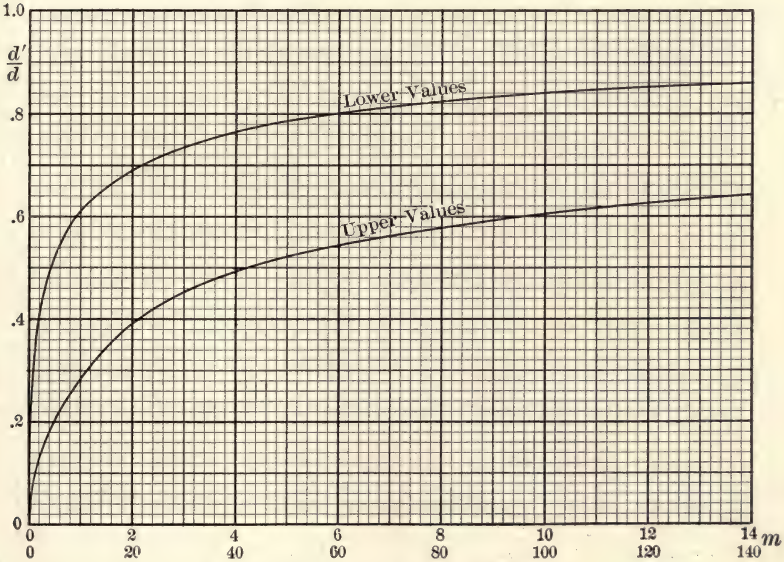


FIG. 186.—Values of m for Gate Valves.

(For Upper Curve use Lower Valves.)

constructed from the reported results of Williams, Hubbel and Fenkell, Weisbach and Grashof. In these R = radius of pipe bend of 90°, d = diameter of pipe, the quantity d' = amount

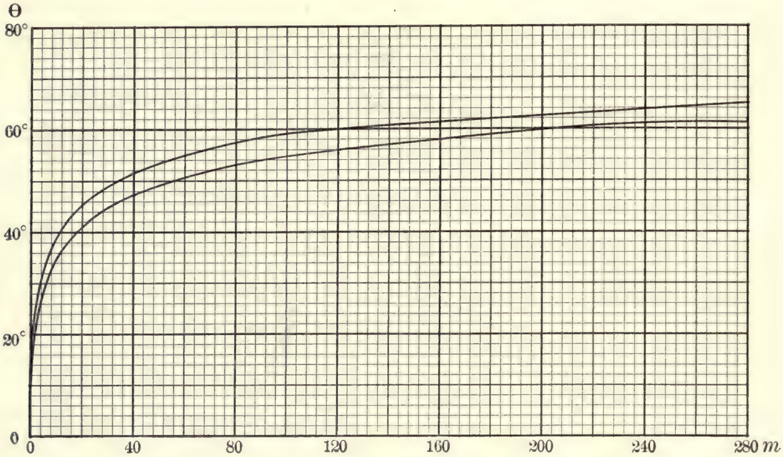


FIG. 187.—Values of m for Cocks and Butterfly Valves.

Upper Curve—Cocks.
Lower Curve—Butterfly Valves.

the valve has been closed, and θ the angle the cock or butterfly valve has been turned. Fig. 188 shows the form of valves used.

The loss due to sudden enlargement which occurs when the

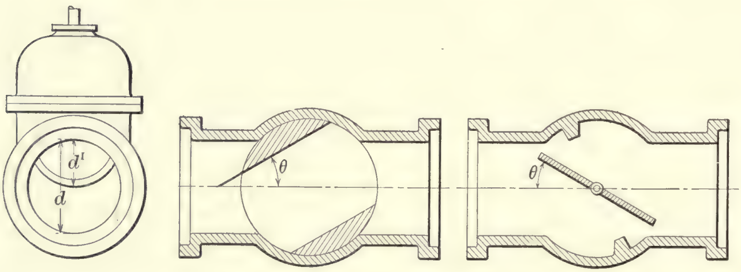


FIG. 188.—Gate Valve, Cock and Butterfly Valve.

water enters the cylinder from the valves or from the suction pipe is equal to

$$h_{en} = \frac{(v_s - v)^2}{2g} = \left(1 - \frac{A_s}{A}\right)^2 \frac{v_s^2}{2g}, \quad . \quad . \quad . \quad . \quad (68)$$

in which v_s is the velocity before enlargement and v the velocity after this occurs.

The loss due to sudden contraction is much less for the same change of section, and is equal to

$$h_{sc}=\left(\frac{1}{k}-1\right)^2\frac{v_s^2}{2g}. \qquad (69)$$

where

$$k=0.582+\frac{0.0418}{1.1-r}, \qquad (70)$$

r = ratio of the diameter of the small pipe to that of the large pipe.

These various equations give the values ζ for each one of the losses, and hence the expression

$$\left(\Sigma\zeta_s\left(\frac{A}{A_1}\right)^2+1\right)\frac{s^2}{2g}$$

may be found. In the problem considered these quantities reduce to the following:

At entrance

$$h=.5\frac{v_s^2}{2g}=.5\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g}=.74\frac{s^2}{2g}.$$

In suction pipe

$$h=f\frac{lv^2}{2gd}.$$

$$\text{Mean } s=2rN'=2\times\tfrac{15}{12}=2.50 \text{ ft. per sec.}$$

For pipes

$$f=\frac{0.026}{(2.5\times 1.5)^{\frac{1}{4}}}=.021;$$

$$h=\frac{.021\times 10}{1.5}\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g}=.21\frac{s^2}{2g}.$$

One bend

$$R=3 \text{ feet};$$

$$h=kf\frac{l}{d}\,\frac{v^2}{2g};$$

$$kf\frac{l}{d}=0.075;$$

$$h=.075\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g}=0.112\frac{s^2}{2g}.$$

Sudden enlargement into suction chamber,

$$h=\left(1-\frac{A_1}{A}\right)^2\frac{v_s^2}{2g}=\left(1-\frac{1.77}{2.16}\right)^2\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g}=0.05\frac{s^2}{2g}.$$

The total term will then be

$$\left(\Sigma\zeta_s\frac{A}{A_1}+1\right)\frac{s^2}{2g}=[0.74+0.21+0.112+0.05+1]\frac{s^2}{2g}=2.122\frac{s^2}{2g}.$$

The terms show the importance of the various friction losses. In the present instance it is seen that the loss at entrance is the most important item.

The next term of the expression for h_s is that involving the acceleration. Let

$$\Sigma\frac{AL_1}{A_1}=\frac{2.16}{1.77}\times 9=10.9,$$

and

$$\Sigma\frac{AL_2}{A_2}=\frac{2.16}{2.16}\times 1=1;$$

$$\therefore \qquad \Sigma\frac{AL_1}{A_1}+\Sigma\frac{AL_2}{A_2}=10.9+1=11.9.$$

This total term may be divided into two parts, one $\frac{Xa}{g}$ and the other $\left(\Sigma\frac{AL_1}{A_1}+\Sigma\frac{AL_2}{A_2}\right)\frac{a}{g}$. The latter equals $\frac{11.9}{32.2}a=.38a$.

The value of Xa is found by multiplying the expression for X and a in Eqs. (2) and (4) or by multiplying the values of X

and a if these have been computed as on page 192, where the value of n is 6.

From the first:

$$Xa = \omega^2 r^2 \left[1 - \cos\theta + \frac{1}{2n}\sin^2\theta\right]\left[\cos\theta + \frac{1}{n}\cos 2\theta\right]$$

$$= \omega^2 r^2 \left[\cos\theta - \cos^2\theta + \frac{1}{2n}\sin^2\theta\cos\theta + \frac{1}{n}\cos 2\theta - \frac{1}{n}\cos\theta\cos 2\theta + \frac{1}{2n^2}\sin^2\theta\cos 2\theta\right].$$

The third and last terms of this could be omitted, but the other terms are of importance; thus the approximate value would be

$$Xa = \omega^2 r^2 \left[\cos\theta(1 - \cos\theta) + \frac{1}{n}\cos 2\theta(1 - \cos\theta)\right]$$

$$= \omega^2 r^2 \left[\cos\theta + \frac{1}{n}\cos 2\theta\right][1 - \cos\theta].$$

From the second for $n = 6$, the values are:

$\theta = 0$	15	30	45	60	75	90
$\frac{xa}{\omega^2 r^2} = 0$	0.044	0.147	0.237	0.235	0.093	-0.181
$\theta =$	105	120	135	150	165	180
$\frac{xa}{\omega^2 r^2} =$	-0.592	-0.911	-1.236	-1.494	-1.621	-1.667

These values of the different terms of Eq. (21) are now plotted in Fig. 189, in which the resultant pressures above the absolute zero are given when $\phi = 90°$.

It is to be remembered in this work that dimensions are in feet.

In Fig. 189 the various curves are plotted for different piston positions. The positions of the piston for the crank positions of 0°, 30°, 60°, 90°, 120°, 150°, and 180° have been shown by vertical lines. The important lines of pressure are marked in different ways. The atmospheric pressure, of zero

gauge pressure, or 34 feet absolute is shown by a dot-and-dash line while the lift h_2 of 10 feet is represented by a dash-and-three-dot line.

The friction loss varies with s^2 and is given by a dash-and-two-dot line. This curve starts from zero at the piston position for 0° and reaches a maximum just before 90°, while the value at 180° is again zero.

The valve friction is represented by a dash and two-dot line and remains practically constant during the larger portion of the stroke according to the experiments of Hartmann-Knoke.

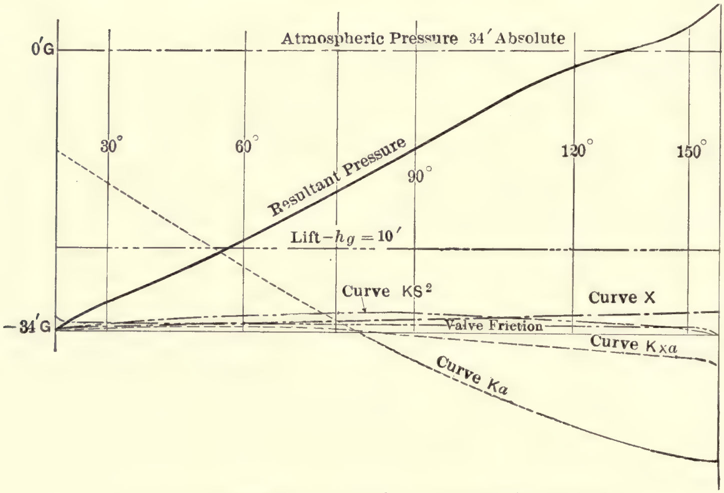


FIG. 189.—Pressure on Suction without Air Chamber.

The position of the inertia effect due to the water in the cylinder has been shown to vary with the product xa. This is represented by a dash line which starts at zero, increases to a maximum about 45°, passes through zero just before 90° and increases negatively to the end of the stroke. This quantity is the same for each end of the cylinder and for each stroke.

The chief disturbing element in the suction pressure is the inertia of the water in the pipe line. This is the quantity given by $Y=ka$ and is shown by the dotted line. The water

in the suction pipe has to have an acceleration proportional to that of the piston if the column remains intact, and consequently to produce this a great head results.

The curve representing the head lost due to piston travel, x, is shown by a slanting dot-and-dash line.

The combined curve is drawn as a solid line. At the beginning of the stroke the gauge pressure is -34 feet, or absolute zero. This means that the column would probably break and produce knocking. The pressure increases rapidly, however, reaching a value above the atmosphere at the end of the stroke. This shows that although the water is lifted 10 feet $+2\frac{1}{2}$ feet, the inertia of the water in the pipe is so great that on trying to stop it the pressure increased to the value shown in the figure.

The pressures marked on this and the other pressure diagrams, Figs. 191, 197 and 198, are measured in feet, starting from atmospheric pressure. The absolute zero of pressure is 34 feet below this.

The absolute pressure of this figure at its lowest point can never be a negative quantity, as there is never a negative absolute pressure. Should the diagram give this in the combination of the various lines this would indicate that the water column would separate and shock would result. This occurs when the pressure assumes an apparent negative value. h_s is therefore limited to zero. This value cannot be obtained, as the pressure of boiling for the water temperature would fix the lowest value. When this pressure is reached the column will separate, since steam will form. This gives then as a limit

$$h_{\text{steam at } t^\circ} < h_a - h_2 - x \sin \phi - h_{sv} - \left(\Sigma \zeta \frac{A}{A_1} + 1\right)\frac{s^2}{2g} - \left(X + \Sigma \frac{A}{A_1} L_1 + \Sigma \frac{A}{A_2} L_2\right)\frac{a}{g}.$$

At the end of the upper stroke when $s = 0$

$$h_{\text{steam at } t^\circ} < h_a - h_2 - l \sin \phi - h_{sv} - \frac{l a_{180}}{g} - \Sigma \frac{A}{A_1} L \frac{a_{180}}{g},$$

or

$$L<\frac{h_a-h_2-l\left(\sin\phi+\dfrac{a_{180}}{g}\right)-h_{st}-h_{sv}}{\Sigma\dfrac{A}{A_1}\,\dfrac{a_{180}}{g}}, \quad . \quad . \quad . \quad . \quad . \quad (72)$$

at 90°

$$L<\frac{h_a-h_2-x_{90}\left(\sin\phi+\dfrac{a_{90}}{g}\right)-h_{sv}-\left(\Sigma\zeta\dfrac{A}{A_1}+1\right)\dfrac{S_{90}}{2g}-h_{st}}{\Sigma\dfrac{A}{A_1}\,\dfrac{a_{90}}{g}}. \quad (73)$$

These give the limiting values of L with which the pump will operate without breaking the column of water or pounding. Should L be made greater the pump will pound.

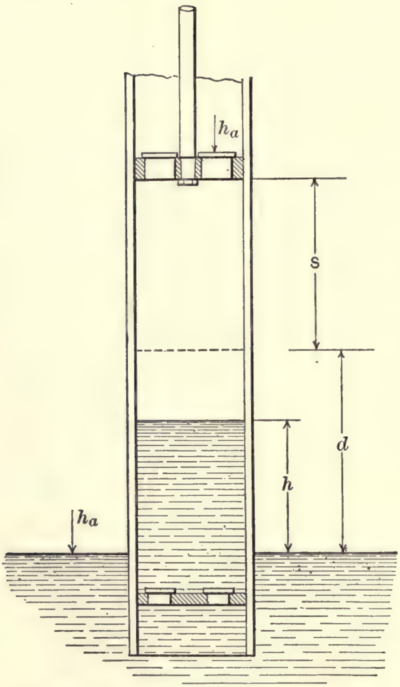


FIG. 190.—Suction Lift.

At times small pumps are built with foot valves at the base of the pump and a bucket plunger in the same barrel at some height above. Such pumps have a limiting distance between the piston and foot valve. Consider the pump (Fig. 190) to have the same bore as the pipe below, and that the water has been sucked in until it reaches the height h, at which time the action stops because when the piston is at its lower end of stroke the pressure of the air is just atmospheric, while at its upper end it is h_a-h. That is, the pressure will not be sufficiently low to allow any more water to enter when at the top of its stroke. The following equation holds then from Boyle's law:

$$(h_a-h)A(s+d-h)=h_aA(d-h);$$
$$(h_a-h)(s+d-h)=h_a(d-h);$$
$$h_as+h_a(d-h)-(s+d)h+h^2=h_a(d-h);$$
$$h^2-h(s+d)=-h_as;$$
$$h=\tfrac{1}{2}(s+d)\pm\sqrt{\tfrac{1}{4}(s+d)^2-h_as}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (74)$$

Now when $\frac{1}{4}(s+d)^2=h_as$ there are two equal roots for h or the water will rise to a certain point.

If $\frac{1}{4}(s+d)^2>h_as$ there are two real roots, which means that the water will rise to a certain point, but if it could be moved beyond this point it would go still further to another point h.

If $\frac{1}{4}(s+d)^2<h_as$ there will be no real roots or the water will be sucked up and the pump will operate.

This gives

$$\tfrac{1}{2}(s+d)<\sqrt{h_as},$$

or

$$d<2\sqrt{h_as}-s. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (75)$$

This then gives the limiting lift for such a pump when the suction valve is placed in the bottom of the pipe.

The same principle is applicable to pumps in which the volume of the cylinder between valves is large compared with the displacement of the pump. Let V_c be the volume of the cylinder between valves when the piston is at the end of the stroke (called clearance on steam cylinders) and V_r the volume displaced by the piston, then if the pump is to operate under the suction lift h_s to the suction valves without filling the pump barrel with water to prime the pump for starting, the following equation must hold:

$$(h_a-h_s)(V_r+V_c)>h_aV_c.$$

Under this condition the air in the cylinder on the stroke at which water just reaches the suction valve will be compressed to a little over atmospheric pressure and be driven out. This gives

$$h_s<\frac{h_aV_r}{V_r+V_c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (76)$$

V_c is really what might be called the clearance volume, and it is seen that when this is zero the limiting suction head is h_a; when it is equal to V_r, h_s becomes $\frac{1}{2}h_a$ at the limit.

The limitations on this deduction must be kept in mind. The value of h_s in formula (76) is the suction head against which a pump with air-tight valves will operate without priming. If the pump is primed, however, this water fills the clearance space and V_c becomes equal to zero, and then the pump will operate against the limiting head h_a if no friction is considered. If friction is considered the values given in Eqs. (72) and (73) are the limiting suction heads.

Passing now to the discharge side it is seen that the expression for the pressure below the piston on the down stroke will be

$$h_d = h_a + h_3 - x \sin \phi + h_{dv} + \left[\Sigma\zeta\left(\frac{A}{A_1}\right)^2 + 1\right]\frac{s^2}{2g}$$
$$+\left(x + \Sigma\frac{A}{A_1}L_1 + \Sigma\frac{A}{A_2}L_2\right)\frac{a}{g}. \quad . \quad . \quad (77)$$

As before the various losses will be plotted in Fig. 191.

To plot these losses for the problem stated throughout this chapter, the following numerical values hold:

$h_a = 34$;

$h_3 = 222$ ft. (10 ft. of the 232 is due to losses);

$x \sin \phi = \mathrm{X}$;

$h_{dv_0} = 226$ ft.;

$h_{dv} = 1.36$;

$$f\frac{l}{d}\left(\frac{A}{A_1}\right)^2 = 0.021 \times \frac{200}{1.5}\left(\frac{2.16}{1.77}\right)^2 = 4.2.$$

Assume 4 bends of size used in suction

$$\zeta_b = 4 \times .075 \times \left(\frac{2.16}{1.77}\right)^2 = 0.5.$$

Assume that all valves are open wide and that loss due to sudden contraction is negligible.

$$\left(\Sigma\left(\frac{A}{A_1}\zeta\right)+1\right)\frac{s^2}{2g}=(4.2+0.5+1)\frac{s^2}{2g}=5.7\frac{s^2}{2g};$$

$$\frac{xa}{g}=Z;$$

$$\left(\Sigma\frac{A}{A_1}L+\Sigma\frac{A_1}{A_2}L_1\right)\frac{a}{g}=W=\left(\frac{2.16}{1.77}\times 200\right)\frac{a}{g}=7.6a.$$

These are now plotted in Fig. 191.

This figure shows the great effect of the inertia of the water, and in working it out the effect of the compression of the water has been neglected, as the use of the air chamber makes

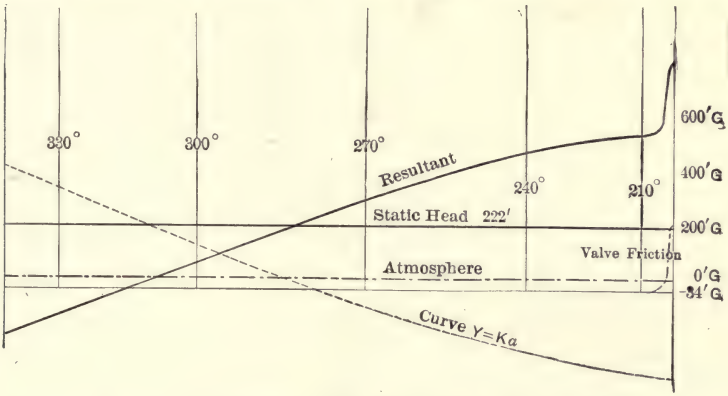


FIG. 191.—Pressure on Discharge without Air Chamber.

such a condition impossible. However, the problem has been computed to show what is eliminated by the use of the air chamber.

The scale of head has been increased so that the heads could be plotted and so great is the scale that the loss, due to friction, the larger part of the valve friction, and the $\frac{xa}{g}=Z$ terms do not show. The important curves are the static head, 256 feet above the absolute zero, the curve due to acceleration of the water in the pipe, $Y=Ka$, and the atmospheric line at 34 feet. These combine with the initial valve friction to give a line which starts with a value of over 800 feet gauge pressure and ends with a pressure below absolute zero. This

last pressure is impossible, and as a result water will separate from the piston or water may be sucked through from the suction side.

The great variation should be eliminated, and one method is to run the pump more slowly or cut down the mass of water which has to be accelerated. This latter is done by the use of the air chamber, as will be shown.

The diagrams seen here and in Figs. 174, 175, 189, 197,

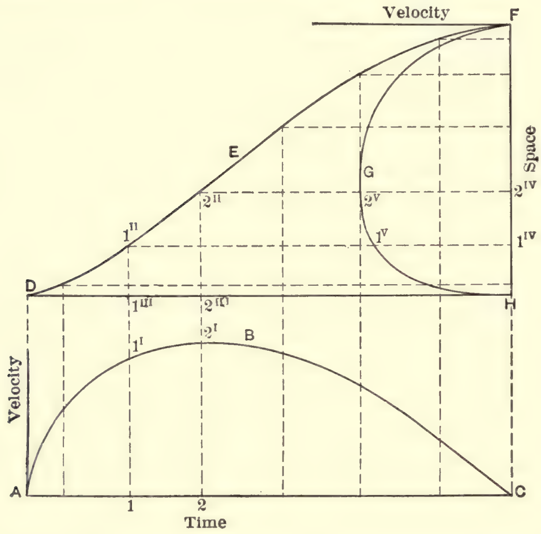


FIG. 192.—Velocity-space from Velocity-time Diagram.

and 198 have been on time and space bases, and it is well to see how a curve on one base may be changed to one on another base.

Fig. 192 gives a velocity diagram *ABC* on a time base. To change this to a space base the curve *DEF* has been drawn with an integraph such that the height $1'' \; 1'''$ directly over the point 1 is proportional to the area $A11'$, $2'' \; 2''' \equiv$ area $A22'$, etc. The planimeter may be used for the determination of these areas and from them the ordinates $1'' \; 1'''$, $2'' \; 2'''$, etc.

Now area $A11' = \int_0^t vdt =$ space. Hence the ordinates of the curve *DEF* are the spaces passed over in the times repre-

sented by the points $1''$ $2''$, etc., or the curve is a time space curve.

If now the horizontal lines $1''$ 1^{iv}, $2''$ 2^{iv} be drawn and the distances 1^{iv} 1^{v}, 2^{iv} 2^{v}, be laid off equal to $1'$ 1, $2'$ 2 respectively, the curve *FGH* is obtained which is the velocity curve on a space base.

The curve in Fig. 193 is a velocity curve on a space base.

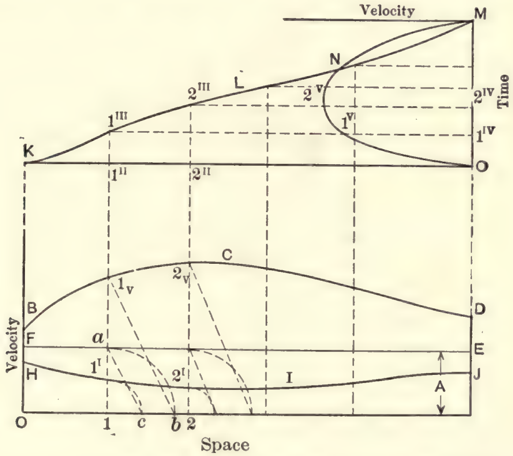


FIG. 193.—Velocity-space Diagram to Velocity Time.

In this the method of getting to a time base is not direct. The following method is used:

$$S=\int vdt;$$

$$ds=vdt;$$

$$dt=\frac{ds}{v};$$

$$t=\int\frac{1}{v}ds.$$

Hence the time taken to go a distance s may be expressed as the area of a curve if the quantity $\frac{1}{v}$ is an ordinate and the space passed over is the abscissa.

The method of procedure is to construct a reciprocal curve

to the one given in which the ordinates are quantities $\frac{1}{v}$ and to draw the integral curve of this.

To get the reciprocal curve of *BCD* construct the line *EF* at any distance *A* from the base and on the ordinate $1, 1_v$ desired swing $1a$ down to b, join 1_v and b and draw ac parallel to 1_vb, and the distance $1c$ will be proportional to the reciprocal of v. This distance is then swung up to 1^1 giving $1, 1^1$ proportional to $\frac{1}{v}$.

This is seen as follows:

$$\frac{1c}{1a}=\frac{1b}{1\ 1_v};$$
$$1c=x;$$
$$1a=A;$$
$$1b=A;$$
$$1\ 1_v=v;$$

hence $$x=A^2\frac{1}{v}.$$

The ordinate of the curve *HIJ* when divided by A^2 gives the reciprocal of v. If A be made unity

$$1\ 1^1=\frac{1}{v}.$$

The integration of the areas under *HIJ* gives the integral curve *KLM* as before, and the velocity curve is constructed by laying off thedistances $1, 1_v, 2, 2_v$ on the proper horizontal lines.

For purposes of studying forces acceleration diagrams are desired. To construct such from various points of the crank or piston movement of a pump when the velocity is known, the following methods are used:

Let Fig. 194 represent the variation of velocity with the space passed over by the body having that velocity.

In this $\frac{dv}{ds}$ = tangent of the angle of inclination of the tangent to the curve with the horizontal.

Now

$$\frac{dv}{ds} \times \frac{ds}{dt} = -\frac{dv}{dt} = a;$$

$$\frac{ds}{dt} = V; \qquad \therefore \; V\frac{dv}{ds} = a. \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

Hence, if V is multiplied by $\frac{dv}{ds}$ the acceleration is obtained.

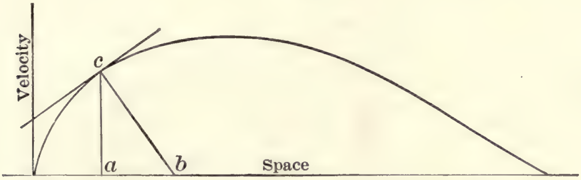


FIG. 194.—Acceleration from Velocity-space Diagram.

$V\frac{dv}{ds} = ab$ if $ac = V$. The subnormal of the sv curve on the s axis represents the acceleration.

If V is on a time base (Fig. 195) the operation is different. In this case $\frac{dv}{dt} = a =$ the tangent of the angle of inclination of the tangent of the curve with the axis of t.

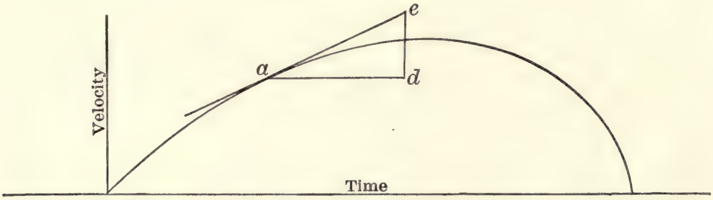


FIG. 195.—Acceleration from Velocity-time Diagram.

If then ad is laid off from the point of tangency equal to unity, ed will equal the acceleration.

Since

$$ed = ad \tan ead = ad\frac{dv}{dt} = 1 \times \frac{dv}{dt} = a.$$

In the case of the figures in Fig. 175, showing the discharge of the water from pumps of various kinds, these are really on a

time base as they represent velocities of the pistons, or quantities of water passing through the pumps per second at any instant, and therefore velocity in the main, for different positions of the crank which is moving at a uniform rate.

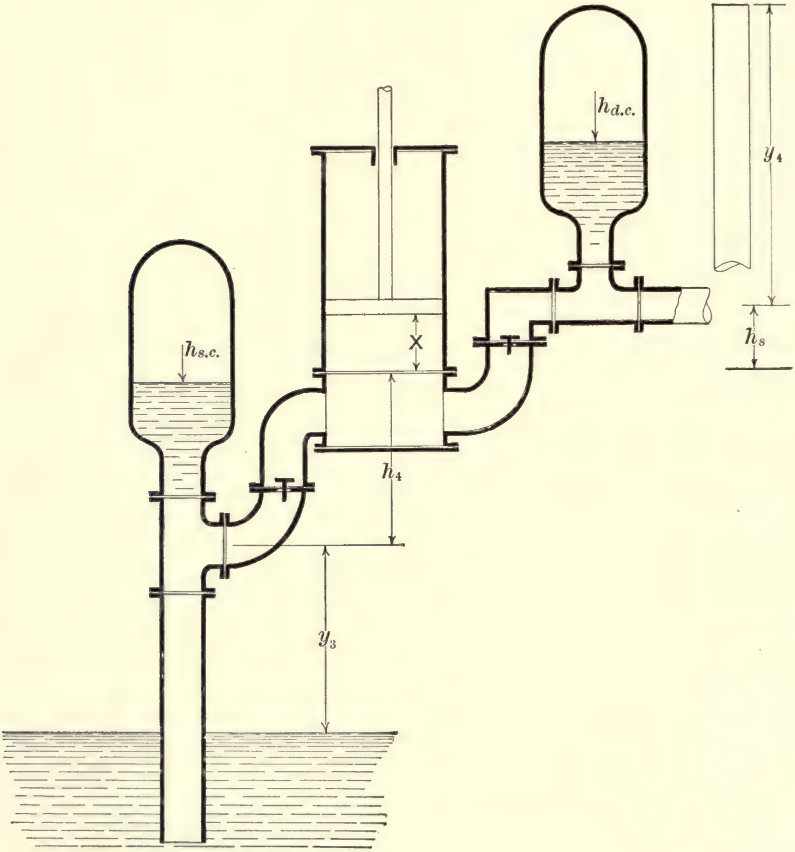


FIG. 196.—Pump with Air Chambers.

The acceleration diagrams could therefore be computed by the second method if desired.

In the figures and in the computation of the h_{sv_0} and h_{dv_0} the effect of inertia in the suction pipe and discharge pipe can readily be seen. These masses of water have to change their velocity as the piston changes its velocity or if the pump is a

multicylinder pump the curves which have been computed for the resultant discharge (Fig. 175) give the velocity variations which exist.

To cut down the variation of the velocity change in the suction and delivery mains and to make the inertia forces smaller, air chambers are introduced on the suction and discharge sides of the pump. The object of the air chamber is attained where so designed that the variation of pressure in it will be slight and hence keep the water in the main under a practically constant velocity while the water which is accelerated is only that which lies between the pump and the air chamber.

The air chambers on the two sides of the pump (Fig. 196) are subject to a certain amount of variation of pressure and in making the preliminary investigation it is well to consider the mean pressure in the air chamber as constant.

This pressure is that within the suction air chamber which causes a flow from the forebay overcoming the resistance of entrance, velocity, friction, and direct head. In the discharge air chamber the pressure is used to give the velocity, lift the water, and overcome friction of all kinds. The pressure in the suction chamber is less than atmospheric pressure in general, although water might be supplied to the pump under pressure from a higher source. The pressure in the discharge chamber is greater than that of the atmosphere. In any case if pressures are always measured from absolute zero

$$h_{sc}=h_a-(\zeta_1+\zeta_2+1)\frac{v_s^2}{2g}-y_3, \quad . \quad . \quad . \quad . \quad (80)$$

$$h_{dc}=h_a+(\zeta_1+\zeta_2+1)\frac{v_d^2}{2g}+y_4. \quad . \quad . \quad . \quad . \quad (81)$$

where the coefficients ζ refer to the losses in the pipes due to friction, and the quantities y are the lifts.

These pressures in the air chambers are now the datum planes from which to compute the pressure in the cylinder beneath the plunger on the two strokes; the water in the con-

necting pipes and cylinder between the piston and the air vessels being the only water which has to be considered in the expression for inertia and friction.

The expressions for pressure now become

$$h_s = h_{sc} - \left[h_4 + x \sin\phi + h_{sv} + \Sigma\zeta\left(\frac{A}{A_1}\right)^2 \frac{s^2}{2g} \right.$$
$$\left. + \left(\frac{s^2 - v_s^2}{2g}\right) + \left(x + \Sigma\frac{A}{A_2}L_4\right)\frac{a}{g}\right]. \quad . \quad (82)$$

The term $\frac{s^2}{2g}$ of Eq. (21) is changed to $\frac{s^2}{2g} - \frac{v_s^2}{2g}$ because $\frac{v_s^2}{2g}$ is included in h_{sc}. The term L_4 is much less than L, and consequently that term is smaller. h_{sv} does not have such a large value as h_{sv_0}, although h_{sv} is the same over a large part of its range.

h_s must be greater than h_{st}, and this inequality will give the limiting suction height or stroke for given conditions, as in Eqs. (72) and (73).

The computations below are made for suction chambers within 2 ft. of the pump.

From Eq. (80)

$$h_{sc} = 34 - \left(.5 + .021 \times \frac{8}{1.5} + .075 + 1\right)\frac{\left(2.5 \times \frac{2.16}{1.77}\right)^2}{64.3} - 8$$
$$= 34 - .24 - 8 = 25.76 \text{ ft.}$$

From Eq. (21) the various parts making up h_s are

$$h_{sc} = 25.76 \text{ ft.};$$
$$h_4 = 2 \text{ ft.};$$
$$x \sin\phi = \mathrm{X};$$
$$h_{sv} = 1.36.$$

The value h_{sv_0} will have to be computed, as the pressure in the air chamber will give a new value of p_s.

$$p_{ls} = \left[25.76 - \frac{4}{32.16}\frac{2.16}{1.77}58\right]62.5 = 17.36 \times 62.5;$$

$$h_{sv_0} = \frac{1}{62.5}\left[17.36 \times 62.5\frac{9.74 - 7.39}{9.74} + \frac{112 \times 0.5}{9.74} + \frac{4.95 \times 112}{9.74} + \frac{112 \times 0.5}{32.16}58\frac{2.16}{7.39 \times 9.74}\right] = 8.3;$$

$$\Sigma\zeta\left(\frac{A}{A_1}\right)^2\left(\frac{s^2}{2g}\right) = .021 \times \frac{2}{1.5}\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g} = .04\frac{s^2}{2g};$$

$$\frac{s^2 - v_s^2}{2g} = \frac{\left[1 - \left(\frac{A}{A_1}\right)^2\right]s^2}{2g} = -.82\frac{s^2}{2g};$$

$$\frac{xa}{g} = Z \text{ as before;}$$

$$\left(\Sigma\frac{A}{A_2}L_4\right)\frac{a}{g} = (1.2 + 1)\frac{a}{g} = \frac{2.2}{32.2}a = .069a.$$

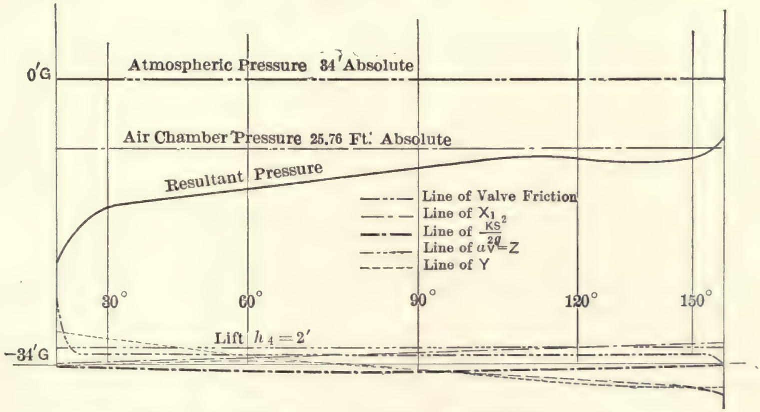


FIG. 197.—Pressure on Suction with Air Chamber.

These values are plotted in Fig. 197.

The pressure on the discharge stroke is given by the equation

$$h_d = h_{dc} + h_5 - x \sin\phi + h_{dv} + \Sigma\zeta\left(\frac{A}{A_1}\right)^2\frac{s^2}{2g} + \frac{s^2}{2g} - \frac{v_d^2}{2g} + \left(x + \Sigma\frac{A}{A_2}L_5\right)\frac{a}{g}. \quad . \quad (83)$$

The computations and curves for the previous problem on the assumption that the air chamber is 4 feet from the pump are given below.

From Eq. (81)

$$h_{dc}=34+\left(.021\frac{19.6}{1.5}+4\times.075+1\right)\frac{\left(2.5\times\frac{2.16}{1.77}\right)^2}{2g}+222$$

$$=34+.23+222=256.23.$$

The parts of Eq. (83) now become

$$h_{dc}=256.23;$$
$$h_5=0;$$
$$x\sin\phi=X;$$
$$h_{dv}=1.36;$$

$$\Sigma\zeta\left(\frac{A}{A_1}\right)^2\left(\frac{s^2}{2g}\right)=.021\times\frac{4}{1.5}\left(\frac{2.16}{1.77}\right)^2\frac{s^2}{2g}=.08\frac{s^2}{2g};$$

$$\frac{s^2}{2g}-\frac{v_d^2}{2g}=\frac{\left[1-\left(\frac{A}{A_1}\right)^2\right]s^2}{2g}=-.82\frac{s^2}{2g};$$

$$x\frac{a}{g}=Z \text{ as before;}$$

$$\Sigma\frac{A}{A_2}L_5\frac{a}{g}=\frac{2.16}{1.77}\times4\frac{a}{32.2}=.152a.$$

The value h_{dv_0} has also to be found when the air chamber is used. In this case

$$h_{ud}=\left[256+\frac{4}{32.16}\,\frac{2.16}{1.77}58\right]62.5=264.4\times62.5;$$

$$h_{dv_0}=\frac{1}{62.5}\left[264.4\times62.5\frac{9.74-7.39}{7.39}+\frac{112\times0.5}{7.39}+\frac{4.95\times112}{7.39}\right.$$
$$\left.+\frac{112\times0.5}{32.16}58\frac{2.16}{7.39^2}\right]=89.8\text{ ft.}$$

In the two quantities h_{sv_0} and h_{dv_0} the effect of the air chamber is seen in the smaller value of the last term, due to a

smaller value of L of the water to be accelerated. The effect of the first term is also to be noted. This has increased the value of h_{sv_0}, while the last term has been changed so much in the expression for h_{dv_0} that the resultant head here has been made smaller.

These values are plotted in Fig. 198.

In Fig. 198 the curves are all marked and from the description of Fig. 189 these may be followed. The two terms of

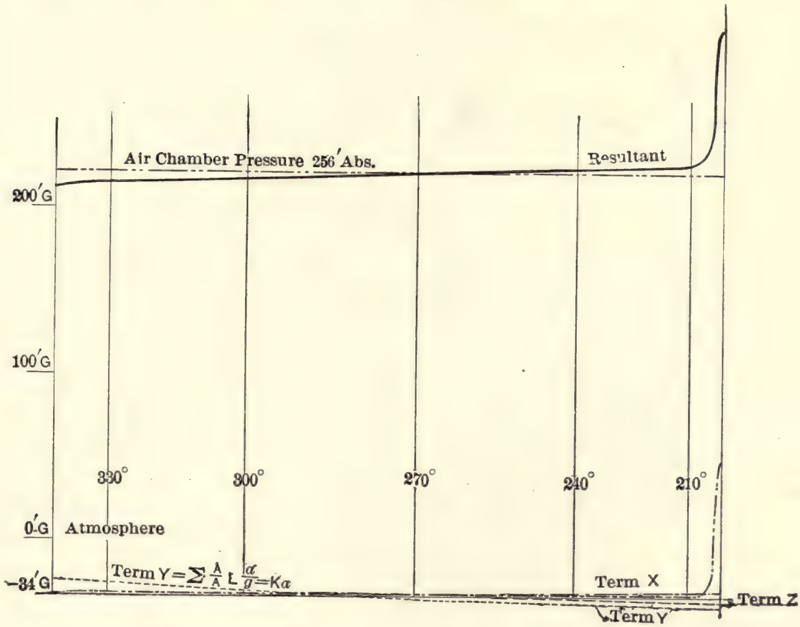


FIG. 198.—Pressure on Discharge with Air Chamber.

Eqs. (82) and (83) involving s^2 have a negative coefficient when combined and consequently this curve is below the line $-34'$ gauge pressure. The effect of the reduction of the mass of the water to be accelerated is seen by the diminished height of the curve $Y=Ka$. The curve $Z=Kax$ and the curve of x are the same as in Fig. 189. The value of the valve friction at the end of the stroke has been increased, although the major portion of the line remains at the same height as before.

By comparing Fig. 197 with Fig. 189 the great advantage

of the air chamber in making the pressure more uniform and in making it possible to draw water at the high speed of the pump is evident. The suction pressure in the pump cylinder never falls to such a low value that the column would break when the air chamber is used.

Fig. 198 shows the action of the air chamber in reducing the Y effect and the friction loss at the opening of the valve at the beginning of the stroke. All of the curves of the various terms are shown except that of the negative term involving s^2. This is so small that it would not be appreciable on the scale of this figure. By comparing this with Fig. 191 the great value of the chamber may be seen.

In Eqs. (80) and (82) the following conditions must hold:

$$h_{sc} > h_{tsc};$$
$$h_s > h_{t_s}.$$

The first of these equations may be used to find the length y_3, and the second to get the limiting length $h_4 + y_3$, or the limiting speed $2LN$ of the pump piston.

The second equation will give the limiting length to the cylinder if the expression for h_{sc} from Eq. (80) is inserted in the expression for h_s.

Size of the Air Chamber

The suction air chamber may be considered to receive water at a steady rate A_sV_s per second, while it gives to the pump an amount which varies for different crank positions as shown by the curves of Fig. 175. The areas of these figures represent quantity handled by the pump so that the area above the mean curve represents the amount supplied by the suction air chamber during a certain part of the stroke while the area below the mean curve represents the amount sent into the air chamber when the pump does not require it. This statement is reversed in the discharge or pressure air chamber when uniform discharge is assumed A_dV_d in the discharge pipe. In this case the area above the line represents the amount given to the air chamber while that below the line represents

the amount given up by the chamber when the pump is giving out less than the quantity discharged through the main.

Calling the amount of water in the air chamber at a, W, the excess areas above mean line ε_1, ε_2, etc., and the deficiencies d_1, d_2, etc., then the quantities at the various points b, c, d, etc., are

$$\left.\begin{array}{ll} \text{At } (a) & W, \\ \text{“} \quad (b) & W \pm \varepsilon_1, \\ \text{“} \quad (c) & W \pm \varepsilon_1 \mp d_1, \\ \text{“} \quad (e) & W \pm \varepsilon_1 \mp d_1 \pm \varepsilon_2, \\ \text{“} \quad (f) & W \pm \varepsilon_1 \mp d_1 \pm \varepsilon_2 \mp d_2. \end{array}\right\} \quad . \quad . \quad . \quad (84)$$

The upper sign would be used for the discharge air chamber and the lower sign for the suction air chamber.

The difference between the greatest and least of these quantities represents the greatest variation of volume of the water in the chamber. Call the difference F.

The volume of the air varies from a maximum to a minimum as this water comes from the pump or suction, so that

$$100\frac{V_{\max} - V_{\min}}{V} = 100\frac{F}{V} = \text{the percentage}$$

variation in volume. Since this compression of air in the chamber is isothermal, $pV = k$. The pressure may be expressed in any units. So the formula will hold for the pressure expressed in feet head, hence

$$h_{sc}V_{sc} = h_{\max}V_{\min} = h_{\min}V_{\max},$$

since the maximum pressure occurs with the minimum volume and vice versa. Solving for $V_{\max}$ and $V_{\min}$ in terms of V_{sc}, and substituting in the equation for the variation of volume there results

$$\% \text{ variation} = \Delta V = \frac{\dfrac{h_{sc}}{h_{\min}}V_{sc} - \dfrac{h_{sc}}{h_{\max}}V_{sc}}{V_{sc}}$$

$$= h_{sc}\frac{h_{\max} - h_{\min}}{h_{\max} \quad h_{\min}}. \quad . \quad . \quad . \quad (85)$$

If $h_{\max}h_{\min}$ is taken as h^2_{sc} which makes h_{sc} a geometric

mean or mean proportional between h_{max} and h_{min}, the value of ΔV becomes

$$\Delta V = \frac{h_{max} - h_{min}}{h_{sc}} = \Delta h,$$

or the percentage variation of pressure is the same as the percentage variation in volume. Hence

$$\Delta h = \frac{F}{V_{sc}},$$

or

$$V_{sc} = \frac{F}{\Delta h}. \qquad (86)$$

If then Δh or the permissible variation in pressure is assumed, the volume of the air chamber to give this result is known for a given F. The equation shows clearly that the air chamber will vary with F and that whenever the resultant piston discharge approaches a mean line, the volume of the air chamber decreases. If the variation Δh is to be made small, the volume V_{sc} will be made large. It is to be noted that Δh is a ratio of variation, and hence if the same ratio of variation is assumed, the volume of the air chamber for a pump will be the same for all pressures or suction lifts. Moreover, according to this the air chamber on the suction side is to be the same as that on the discharge side if the same percentage variation is used, the actual variation being much less on the suction side. This is as it should be, for the pressure change on the suction side is most important, as there is only the atmospheric pressure to do the driving.

The size of the complete air chamber should be such that when the pump is shut down the increase of volume due to the decrease of pressure from the elimination of friction, since the water is at rest, will not be sufficient to cause the water to be entirely driven from the air chamber. The value V_{dc}, computed previously, is the volume of the air in the chamber when it is under the pressure of operation. Calling V'_{dc} the volume of the air when the pump is at rest

$$V'_{dc} = V_{dc} \frac{h_{ds}}{h_d}. \qquad (87)$$

Where h_d is the head when the pump is stationary, and h_{ds} is that while the pump is operating. This volume V'_{dc} is then the minimum volume to be given to the air chamber so that it will never lose the air when the pump is brought to rest.

Although this simple method could be used, Hartmann and Knoke consider the discharge air chamber from the standpoint of supply from the pump when the pump is started up.

Considering the pipe leading to the air chamber to be of area A_c, in which the velocity is assumed to be C_c, it may be said that the quantity entering the air chamber per second is A_cC_c cubic feet. The amount of water leaving the air chamber per second is A_dC_d cubic feet, where A_d is the area of the discharge pipe and C the mean velocity at any instant in the cross-section.

If now C_c be taken as constant the net amount of water entering the air chamber in any time t after the pump has started is

$$q=A_cC_ct-\int_0^t A_dC_d dt. \quad \ldots\ldots \quad (88)$$

It is assumed that it takes this time to get the pump into uniform motion and at the end of the time that the air chamber is giving back as much water as it receives.

The air in the air chamber is originally under the pressure $h_{ds}=h_a+h_d$ in feet of water where h_a is the atmospheric pressure, and h_d is the vertical distance from the end of the pipe to the water level in the air chamber. As this quantity q enters, the total pressure becomes greater, say h_{d_2} due to compression, so far as the chamber is concerned, but produced in reality by the inertia and friction of the water in the discharge pipe. Then if V_{dc} is the volume of the air chamber

$$V_{dc}h_{ds}=(V_{dc}-q)h_{d_2}, \quad \ldots\ldots \quad (90)$$

because the temperature of the air may be assumed constant.

$$q = V_{dc} - V_{dc}\frac{h_{ds}}{h_{d_2}};$$

$$\therefore \qquad dq = V_{dc}h_{ds}\frac{dh}{h^2},$$

if h_{d_2} is considered variable.

Now from (88),

$$dq = A_cC_cdt - A_dC_ddt.$$

Hence

$$(A_cC_c - A_dC_d)dt = V_{dc}h_{dc}\frac{dh}{h^2}. \quad . \quad . \quad . \quad . \quad (91)$$

The pressure in the air chamber at a certain instant is h, while that at the entrance to the air chamber due to the static head is h_0, and there is an unbalanced pressure of $A_d(h - h_{ds})w$, which may be utilized.

This acts on the water in the pipe line l_d, of weight l_dA_dw, and produces the acceleration $\frac{dc_d}{dt}$. The force of inertia is $\frac{1}{g}l_dA_dw\frac{dc_d}{dt}$ expressed in pounds.

Hence

$$A_d(h - h_1)w = \frac{1}{g}l_dA_dw\frac{dc_d}{dt};$$

$$dt = \frac{l_d}{g}\frac{dc_d}{h - h_1},$$

or

$$(A_cC_c - A_dC_d)\frac{l_d}{g}\frac{dc}{h - h_1} = V_{dc}h_{ds}\frac{dh}{h^2};$$

$$\frac{l_d}{g}(A_cC_c - A_dC_d)dc = V_{dc}h_{ds}\frac{(h - h_1)dh}{h^2};$$

$$\frac{l_d}{g}\left[A_cC_cC - A_d\frac{C^2}{2}\right]_0^{C_d} = V_{dc}h_1\left[\log h + \frac{h_1}{h}\right]_{hd_0}^{hd}; \quad . \quad . \quad (92)$$

where $C = C_d$ at the time $h = h_d$.

$A_dC_d = A_cC_c$ because at the instant when the greatest com-

pression occurs in the air chamber there is no net flow into the chamber,

$$\therefore \qquad C_d = \frac{A_c C_c}{A_d}.$$

Then

$$\frac{l_d}{g}\frac{(A_c C_c)^2}{2A_d} = V_{d_c} h_{d_0}\left[\log \frac{h_d}{h_{d_0}} + \frac{h_{d_0}}{h_{max}} - 1\right];$$

or

$$V_{d_c} = \frac{l_d}{h_{d_0}}\frac{(A_c C_c)^2}{2gA_d}\frac{1}{\left[\log \frac{h_d}{h_{d_0}} + \frac{h_{d_0}}{h_{max}} - 1\right]}$$

$$= \frac{l_d}{h_{d_0}}\frac{Q^2}{2gA_d}\frac{1}{\left[\log \frac{h_d}{h_{d_0}} + \frac{h_{d_0}}{h_{max}} - 1\right]}. \quad . \quad (93)$$

To illustrate the method used in the design of air chambers, the size of chamber for the pump will be computed. The pump will be assumed to be double acting, and of the dimensions given before. The curves of Case 2, Fig. 175, represent the action of this pump, and by using a planimeter on the original drawing the right-hand excess area was found to be 0.42 square inch, the left 0.39, the right- and left-end deficiency areas which go together 0.38 square inch, and the middle deficiency 0.43. This gives the following variation in the quantity of water in the air chamber:

At 1, v;
" 2, $v+0.39$;
" 3, $v+0.039-0.43=v-0.04$;
" 4, $v-0.04\ \ +0.42=v+0.38$;
" 1, $v+0.38\ \ -0.38=v$.

The greatest variation is from $v-0.04$ to $v+0.39$, or 0.43 square inch of diagram area.

The figure was originally drawn 6 inches long and five-eighths inch for the mean height. The length represented the time of one revolution or the angle turned through, and the quantity discharged by the pump was 10.85 cubic feet per second. The

scales of the figure are therefore one-sixth second per inch of length and 17.38 cubic feet per second per inch of height. The area scale is therefore 2.89 cubic feet per square inch.

The change in the volume of water in the air chamber is therefore

$$0.43 \times 2.89 = 1.24 \text{ cu.ft.}$$

The variation of pressure Δh is now assumed to be 5 per cent and the volume of the air in the chamber is, by Eq. (86),

$$v_{sc} = \frac{1.24}{.05} = 25 \text{ cu.ft.}$$

To permit the pump to be shut down the air in the discharge chamber on expanding from 256 feet pressure (the pressure in the discharge chamber during action) to 222 feet pressure (static head), the volume of the air vessel should be such that this air is not driven out. By Eq. (87)

$$V'_{dc} = 25\frac{256}{222} = 28.9.$$

The volume of the cylinder is 5.5 cubic feet, and the volume of the air chamber above is about five times this. The vessel would be 30 inches diameter by 48 inches length. The great size of this is due to the kind of pump [Term F of Eq. (86)], and the allowable variation in pressure.

The method of design sometimes employed is to assume the ratio of chamber volume to cylinder volume, and use this only. The ratios suggested are: 3 and 6 times the cylinder volume for single-acting pumps, and one-half to two-thirds of this for double-acting pumps.

From the above formulæ the following steps may be taken to determine the leading dimension of the water system when a given quantity of water, Q per second, is to be pumped:

Size of Pipes and Air Chambers

The suction pipe should be large and as free from bends as possible. In the first approximation a velocity of 3 feet per second may be assumed for the velocity in the suction pipe.

Then

$$A_s = \frac{Q}{v_s}.$$

The size of the suction air chamber is given by Eqs. (86) and (87).

The discharge pipe is in many cases so long that it will pay to compute several sizes. Suppose a pipe is found in which the lost head due to friction is h_f feet and by using a larger pipe this may be reduced to h'_f feet. The gain by this enlargement is

$$\frac{(h_f - h'_f)wQ}{550 \times \text{eff.}} = \text{H.P.}$$

If now the power costs M dollars per horse power hour the saving per year of T hours will be

$$\text{H.P.} \times M \times T.$$

This same could be used to pay the difference in yearly cost between the cost of the small and large pipes. The increased cost of iron and installation could then be found, and if the interest, depreciation, taxes, and insurance on this cost is just equal to the amount saved per year, there would be no economic advantage in putting in the larger pipe. If the amount is less than the saving, a still larger pipe should be tried, while, if the saving is less, it would be well to make an investigation with a smaller pipe to see if the gain in interest, depreciation, taxes, and insurance of a smaller pipe would not be greater than the increased cost of power. This does not consider the development for future service, which would alter the problem.

Having the area A_d of the discharge pipe, its length, and the bends in it, the resistance from such a line can be found, and from it the size of the air chamber for the discharge. Eqs. (86) and (87) are used for this.

Number of Revolutions

Ratio of L to D

These two quantities are mutually dependent. In many cases the quantity $2LN$ or piston speed is the quantity assumed, and then D is known from the formula

$$D=\sqrt{\frac{Q}{(2LN)\frac{n}{2}\frac{\pi}{4}}},$$

where n is the number of active strokes in one revolution.

With $2LN=S$ as soon as N is known L is given by

$$L=\frac{S}{2N}.$$

The speed of the piston, S, is not a constant, but may vary from 50 to 700 feet per minute. The equation shows the common tendency, however, to cut down the stroke as the number of revolutions increases. The piston speeds used with steam engines vary from about 350 to 800 feet per minute, and where the pump piston is mounted in tandem with the steam piston, the piston speed for the pump would fix that for the engine.

The length of the machine may determine the stroke to be used. Where a short machine is desired for any given reason, whether the pump be horizontal or vertical, a short stroke is chosen.

This choice of a short stroke, however, may mean a large diameter and with it much heavier parts, cylinders, piston rods, cross-heads, connecting rods, pins, and other parts. No general rule can be given; each problem arising must have its own solution.

The value of N is determined by many things. Until within the last twenty years pumps were usually run at from 20 to 80 revolutions per minute, and at the higher speeds trouble was experienced. In later years by increasing the valve area or the number of valves, and in cases using positively

actuated valves, by giving large passages and cylinder space, and by the use of large air chambers much higher speeds of revolution were used. As was mentioned earlier, Riedler was one of the first to design high rotative speed pumps. The speeds have been carried up to 250 R.P.M. By using such speeds it has become possible to operate pumps by direct connection to electric motors, and gas engines without the use of intermediate gears. Such arrangements save room, although on account of the special design for such pumps the cost may not be reduced.

In America the practice is to operate large pumps with piston speeds of about 500 feet per minute, and a rotative speed of 25 R.P.M. This selection has its advantage, as the inertia forces vary as the square of the number of revolutions, and as the first power of the crank radius or stroke. With smaller pumps 60 R.P.M. to 80 R.P.M. is often found, and with special pumps the higher speeds of 150 to 200 R.P.M. are used. The latter express pumps are of value where direct connection is necessary, and space or weight limited.

Having then the size of the cylinders, revolutions, size of suction and discharge pipes, the number of valves should be determined, and then the capacity of the air chamber.

CHAPTER VI

DESIGN OF PARTS

Water Cylinders

After the diameter and stroke have been determined, the cylinder is designed. The arrangements of the bore, the passages leading to the valves, the valves, the valve decks, and the valve chests are all of importance, and are varied according to the peculiarities of the designer. The general principle is to make the path of the water as direct as possible; the construction of the casting simple, and the position of the valves such that they may be easily examined and replaced. To show the arrangements proposed and used by pump designers a number of typical pump cylinders have been chosen, each illustrating some special point. Although a few might have answered the purpose, the number has been increased to familiarize the student with the construction of modern pumps, to show many ways of constructing machines, and to illustrate how constructions may be simplified.

The water pistons take one of four principal forms: 1st, the piston (*A*, Fig. 199); 2d, the plunger and ring (*B*, Fig. 199); 3d, the packed plunger (*C*, Fig. 199), and 4th, the bucket (*D*, Fig. 199).

These are packed in different ways. Fig. 199*A* shows a canvas packing inserted on a ledge formed on the piston casting. The packing is usually made of layers of canvas or square-woven cotton indurated with rubber, or it may be a square flax packing. The form of packing shown at *B* has been used for many years; the long sleeve makes practically a water-tight joint, and for clear water this lasts for some time. The sleeve may be renewed when necessary. The plunger (*C*, Fig. 199) is packed on the inside at the center. Such packing

may be placed at the outside. The form is identically the same in both cases. The kind of packing used is the same as that employed on the piston. The cup-leather packing (*D*, Fig. 199) is used with deep-well pumps and with high-pressure pumps. The cup leather forms a good packing in such cases, as the

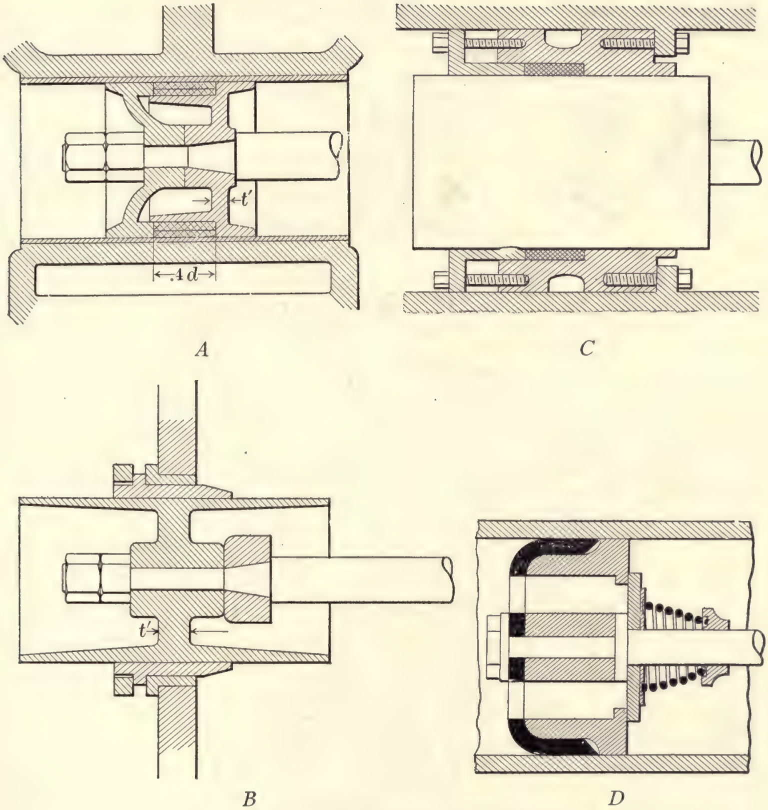


Fig. 199.—Pistons, Plungers and Bucket.

pressure exerted by the leather varies with the water pressure, and so the friction on the suction stroke is small. Fig. 200 shows the form for a double-acting piston with cup-leather packing. When packings are used for the outside they are practically the same in form. The piston rod is packed as shown in Fig. 201 or Fig. 202; the plunger for ordinary pres-

sures with a light packing (Fig. 203), while for heavy pressures in pumps or hydraulic jacks U-leather or hydraulic packing (Fig. 204) is used.

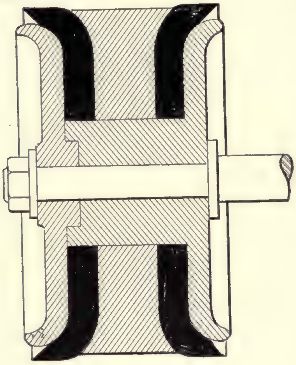


FIG. 200.—Double-leather Packing.

The design of these packings is usually empirical, and the figures shown are marked with numbers which are given in terms of a unit.

In Figs. 201 and 202 the proportional unit is $\frac{1}{4}d+\frac{1}{4}''$ where d = diameter of piston rod.

In Fig. 203 the size and number of the bolts and the thickness of the stuffing box will have to be designed. The thickness A of the packing and the depth B depend to a certain extent on the size of the plunger. $B=0.2$ to $0.4d$ and $A=0.1d+\frac{1}{4}''$ up to $1''$ may be used as a guide.

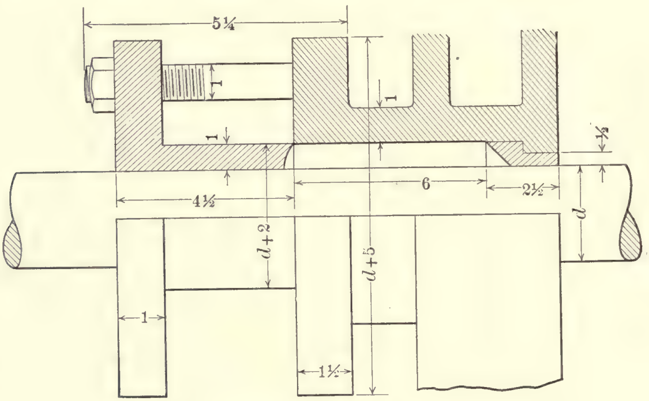


FIG. 201.—Stuffing Box.

The thickness C is designed for high pressures as a cylinder wall under the pressure of the fluid.

The bolt area, a, or the number of bolts, n, may be found by the following formula if one of these quantities is assumed:

$$na=\frac{p\frac{\pi}{4}[(d+2A)^2-d^2]}{S_t}.$$

The cup leathers (Fig. 206) are usually made by soaking the leather to make it pliable, and then forcing it into the mold (Fig. 206) by a press or bolt. After is it forced down

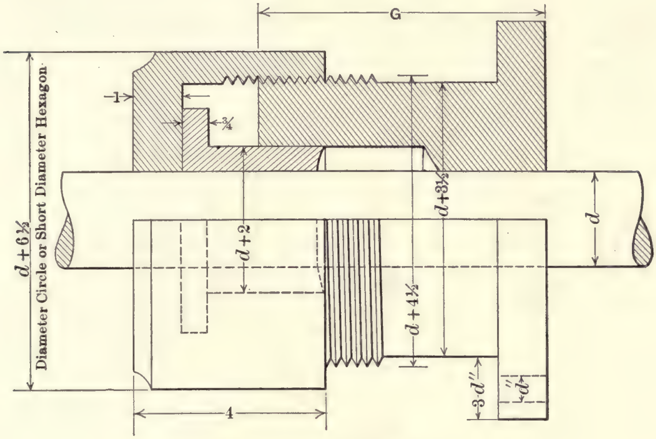


FIG. 202.—Stuffing Box.

it is allowed to dry and set, when it is trimmed on the edge by turning. The wear on cup and other leather packing occurs near the bend in the leather (Fig. 205) as it is this point which

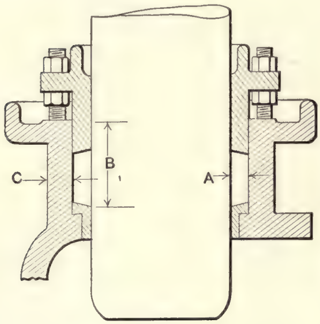


FIG. 203.—Plunger Packing.

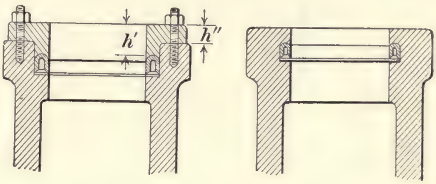


FIG. 204.—U-Leather Packing.

is driven against the plunger or cylinder wall. Since this is the case there is no necessity for making the dimensions h greater than the amount given below.

The U-leathers are proportioned, as shown in Fig. 207, for

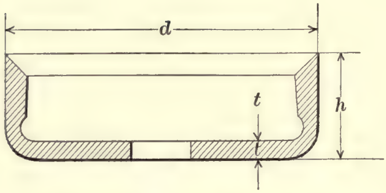


FIG. 205.—Wear in Cup Leather.

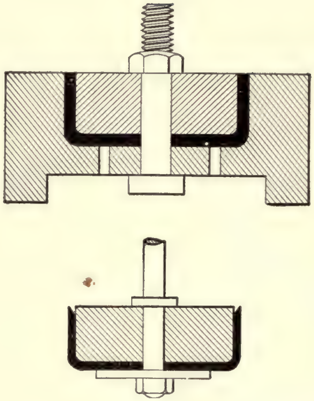


FIG. 206.—Formation of Cup Leather.

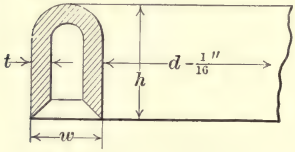


FIG. 207.—U Leather.

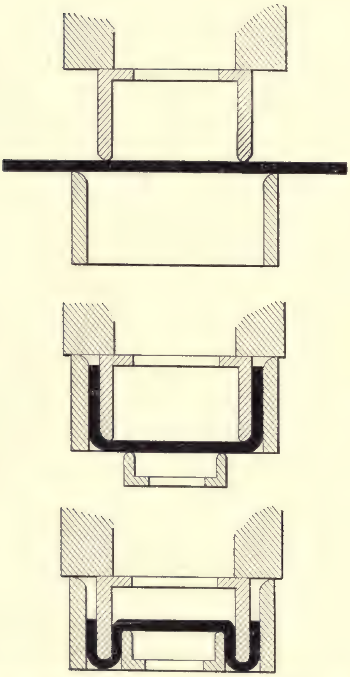


FIG. 208.—Formation of U Leather.

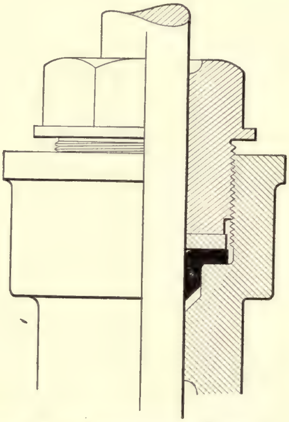


FIG. 209.—Hat-leather Packing.

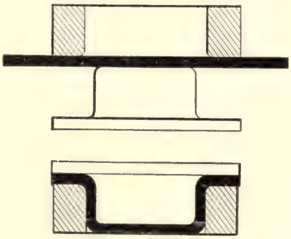


FIG. 210.—Formation of Hat Leather.

which the various dimensions are given later. These are made by formers or molds, after soaking, as shown in Fig. 208. This operation may best be performed on a hydraulic press. The first fold is made as in the second part of the figure, after which the last core is introduced.

The hat leather used as shown in Fig. 209 is formed in the same manner as the cup and U leathers by formers (Fig. 210). After drying, the center is cut out and the edge chamfered.

For cup leathers (Fig. 205),

$$h = 1'' \text{ to } 1\tfrac{1}{2}'';$$
$$t = 0.156 d^{0.28};$$
$$d' = d + \tfrac{1}{16}'';$$
$$\text{Friction} = cdp;$$
$$c = 0.03 \text{ to } 0.05;$$
$$p = \text{hydrostatic pressure per sq. in.}$$

For U leathers (Figs. 204, and 207),

$$\begin{array}{ll}
\text{Diameter box} & = \text{diameter plunger} + 2w - \tfrac{1}{8}''; \\
\text{Outside diameter leather} & = \text{diameter plunger} + 2w - \tfrac{1}{16}''; \\
\text{Inside diameter leather} & = \text{diameter plunger} + \tfrac{1}{16}''; \\
\text{Height} & h = 1.6 \text{ width} = \tfrac{3}{4}'' \text{ to } 1\tfrac{1}{4}''; \\
\text{Width} & = w = \tfrac{1}{2}'' \text{ to } \tfrac{3}{4}''; \\
\text{Bolt area} & = a = \dfrac{\frac{\pi}{4}[(D+2w)^2 - D^2]p}{nS_t}; \\
\text{Height of flange} & h'' = 1.5 \text{ diameter bolts}; \\
 & h' = 3 \text{ diameter bolts}; \\
\text{Width of cylinder flange} & = 3 \text{ diameter bolts}; \\
\text{Friction} & = 0.04\dfrac{P}{d} = 0.03dp.
\end{array}$$

The Marsh pump (Fig. 211) has one of the simplest forms of water end. In this design there is no lining to the cylinder bore. The suction chamber A is formed by coring, and the discharge chamber B is formed by the valve deck plate C and the cover D. The valve seats are forced into holes on the valve decks. The piston is packed with cup leathers. This

particular pump is built for pumping milk and for that reason it is so constructed that it may be taken apart quickly and easily for cleaning. All nuts are wing nuts, and the bolts are hinged so that they may be swung out. The piston rod is light and the stuffing box is simple.

Another form of Marsh pump (Fig. 212) is so constructed that the cylinder bore is fitted with a brass liner to take the wear of the piston. The liner is held in place by screws passing through a flange. It may be renewed easily when necessary. The piston body *F* is fastened to the piston rod by a nut, while

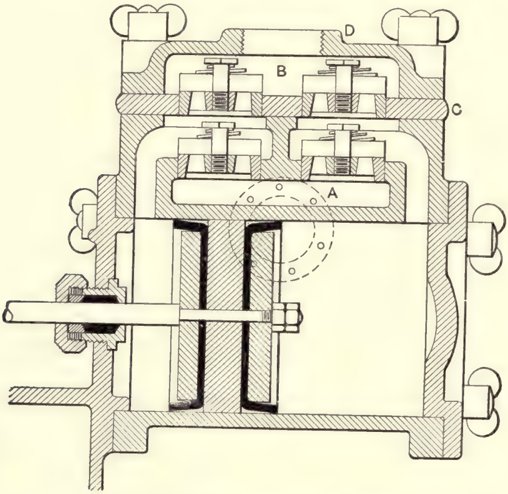


FIG. 211.—Marsh Milk Pump.

the follower plate *G* is held on the same thread by another nut. The removal of this plate permits one to examine or repair the piston packing. The shoulder on the piston rod holds the body *F* in place, preventing any play.

The stuffing box *H* is of the cap form, and is made separate from the cylinder casting. This simplifies the foundry work and lessens the difficulty of the machine work in the manufacture.

The method of dividing the valve chambers of the cylinder ends by a partition carried to the discharge valve deck as well as the method of attaching the air chamber, and the form of

the air chamber are all to be noted. The cap on the end of the air chamber has been employed so that a core print may be used at each end of the pattern.

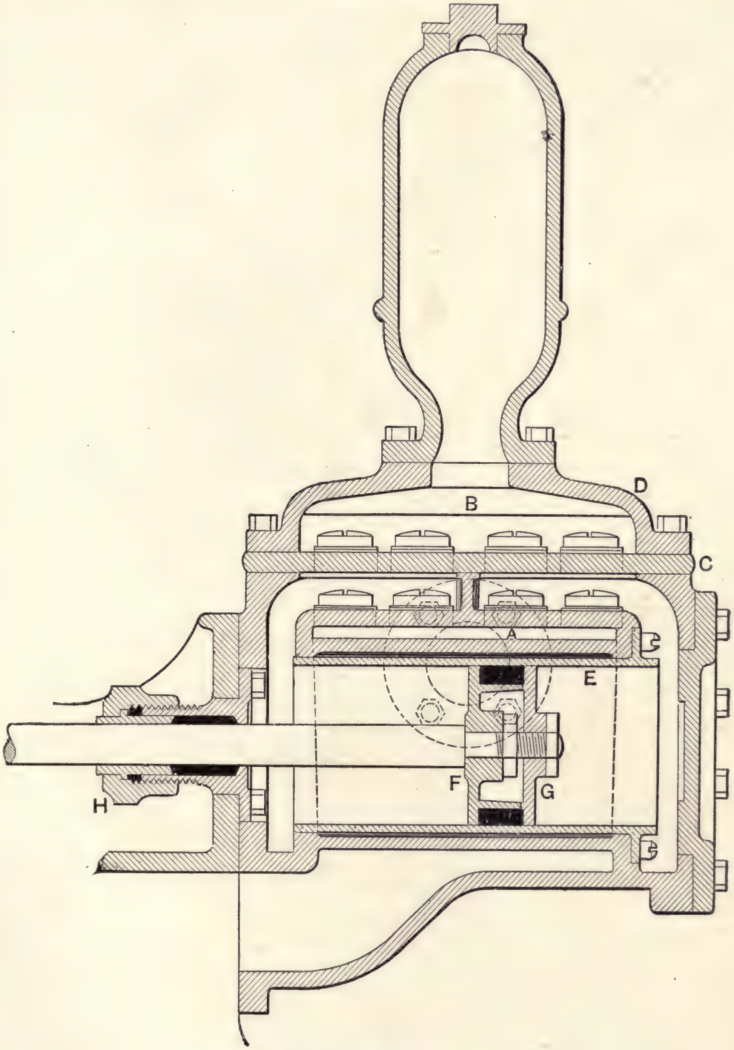


FIG. 212.—Marsh Pump.

To simplify the water end, the Marsh Company build a pump (Fig. 213) in which the water-cylinder bore is made by

using a piece of solid-drawn brass pipe. This makes a very simple casting, as the suction chamber *A* is carried up from the base of the pump. The cap forming the discharge chamber

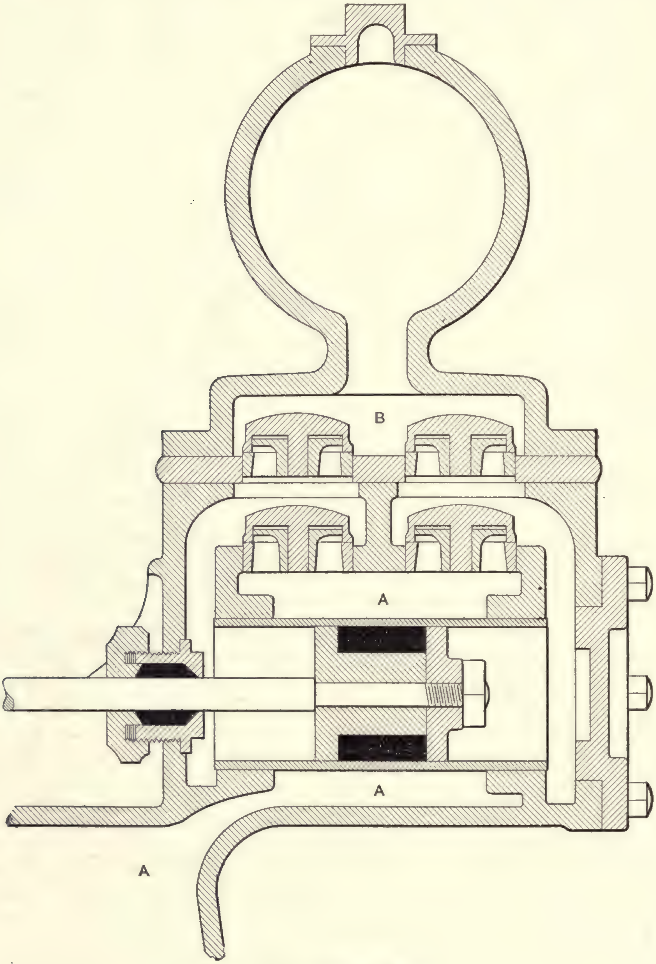


FIG. 213.—Marsh Pump.

B is cast solid with the air chamber. The valves used in this pump are peculiar in form and the guides are so arranged that the valves are not subject to a lifting pressure after they rise to a certain height. The valve seats are inserted in openings

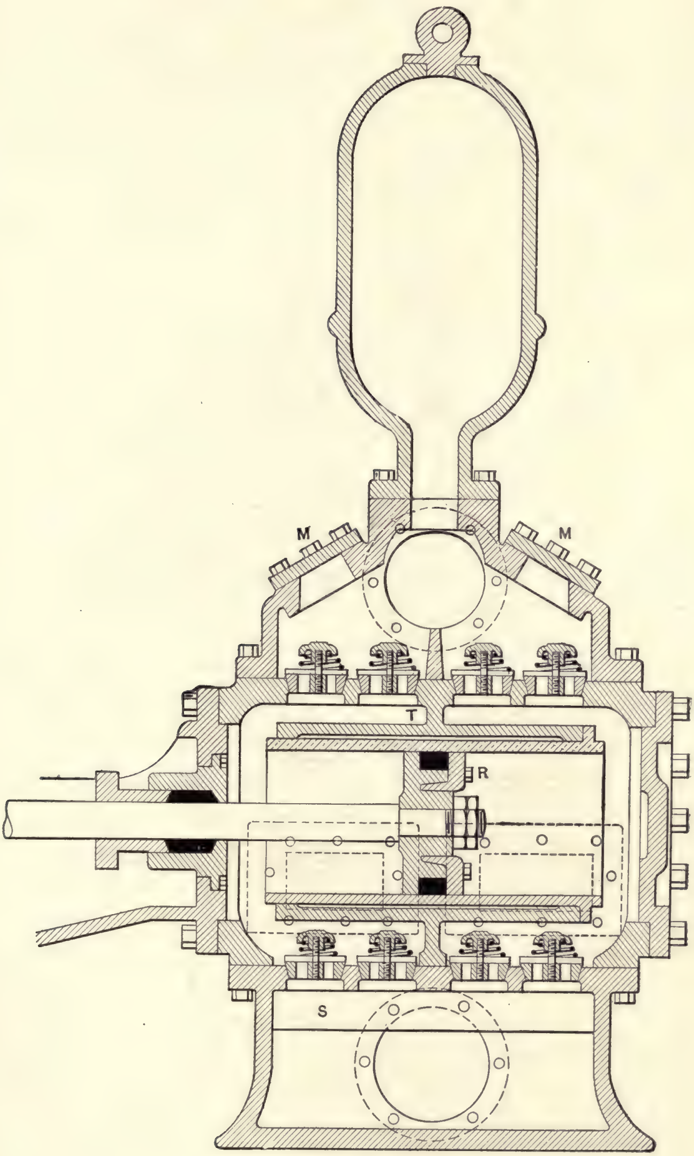


FIG. 214.—Marsh Pump.

in the valve decks, simplifying the casting and making repairing easy. The peculiar form of packed piston, the stuffing box,

and the simple form of casting are to be observed in the figure.

For larger sizes the Marsh pump water end is built as shown in Fig. 214. In this case the suction chamber is a separate casting, and forms the base of the water end. The suction valve deck is reinforced against the water pressure from the cylinder by a rib *S*. Such a construction is necessary, as the valve deck is a broad, flat plate, subject to a downward pressure. A large suction pipe enters the side of the suction chamber. The discharge valve deck is part of the cylinder proper, and is stiffened by the rib *T* which extends across the casting, while the discharge chamber is bolted to the cylinder. The air chamber is attached to this and is similar in form to those used on the smaller pumps. Hand holes are arranged in the side of the cylinder casting, and in the discharge chamber at *MM*, so that the valves may be inspected or repaired without the necessity of removing the large castings or opening large covers. Such construction is always necessary in pumps of any size.

The valves are of a different form from those shown before. These are spring-controlled valves. Their action is similar to that of the others illustrated. The seats are made separate, a good construction, in order that seats may be renewed if broken, and, moreover, the casting is thereby simplified. The follower ring *R* on the piston replaces the plate used on the other pistons, and the bolted gland in the stuffing box replaces the cap gland. The liner of brass is used to simplify renewal when the wear from the piston packing becomes excessive.

One of the important features in pump design is to have an ample and simple direct passage for the water through the pump. The pumps shown in the preceding figures have been good in this respect, but the Fairbanks-Morse pump in Fig. 215 shows in a somewhat better manner, the ample passages. The suction enters the chamber *A* in an easy sweep from *D*, and then goes through a large passage to the pump cylinder. The discharge from the other side is forced into *B* and leaves

through a passage to the discharge E. The general features of this design are seen on inspection of Fig. 215.

Another pattern with large passages is shown in the Worthington pump (Fig. 216). In both of these pumps the cylinder bore is lined by using a piece of brass tubing. This tubing is withdrawn when worn, and another piece inserted in its place. In pumps intended to lift acids which may attack the metal,

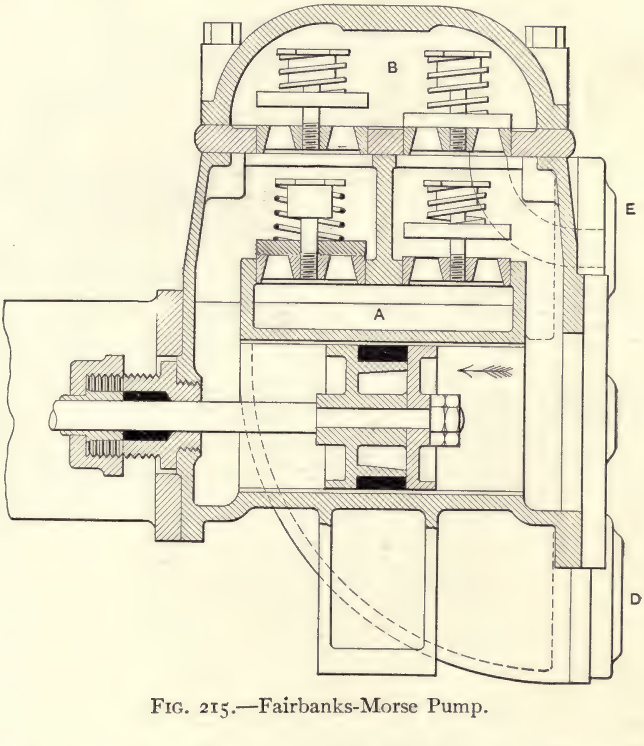


Fig. 215.—Fairbanks-Morse Pump.

a lining of wood has been used for the bore of the cylinder, and the valve chambers as well as the piping.

In Fig. 216 the peculiar form of follower ring or plate is to be noted. This gives a very long wearing surface.

The Worthington plunger and ring form of water end (Fig. 217) is one which has many advantages. The water enters the suction chamber A through C and passes up into the pump cylinders, and from there it is forced into the discharge pipe

D through the chamber *B*. The plunger is fastened to the piston rod by two nuts. It operates through a sleeve *E*, which is held in the partition between the sides by a ring *F*. The sleeve is called a ring. There is no tight packing. The joint

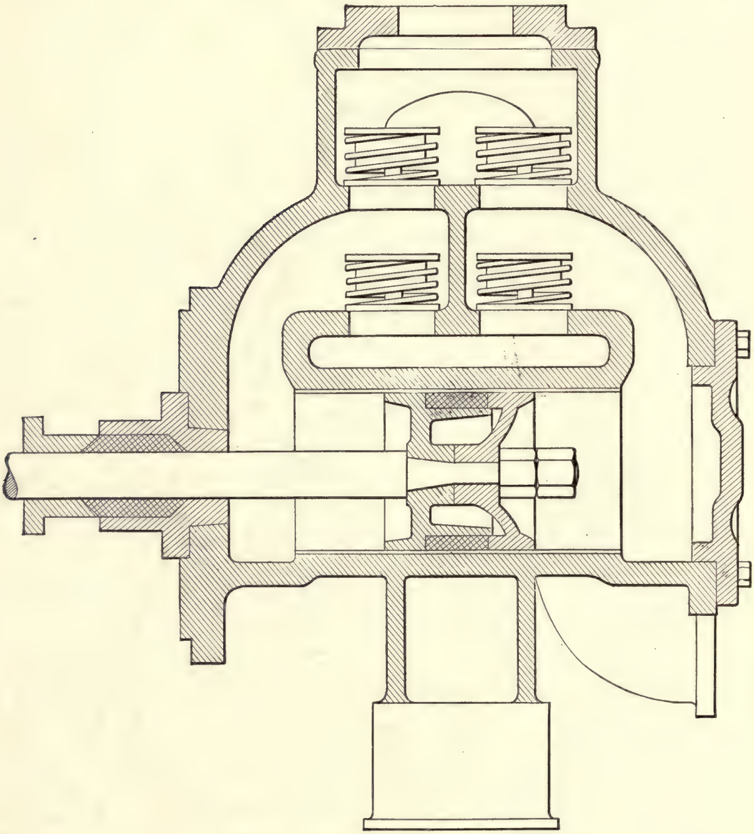


FIG. 216.—Worthington Pump.

is so long that there is practically no leakage. This type of pump has been quite successful.

Outside-packed plunger pumps have the advantage that the leakage past their displacing parts is visible. They are of various forms. The piston rod is at times carried through the two plungers (Fig. 218), the rod passing through a sleeve or stuffing box. In this pump the suction enters at *A* and is

carried to the valves, and finally discharges into the chamber *C* and from it into *B*. This type of end outside-packed plunger

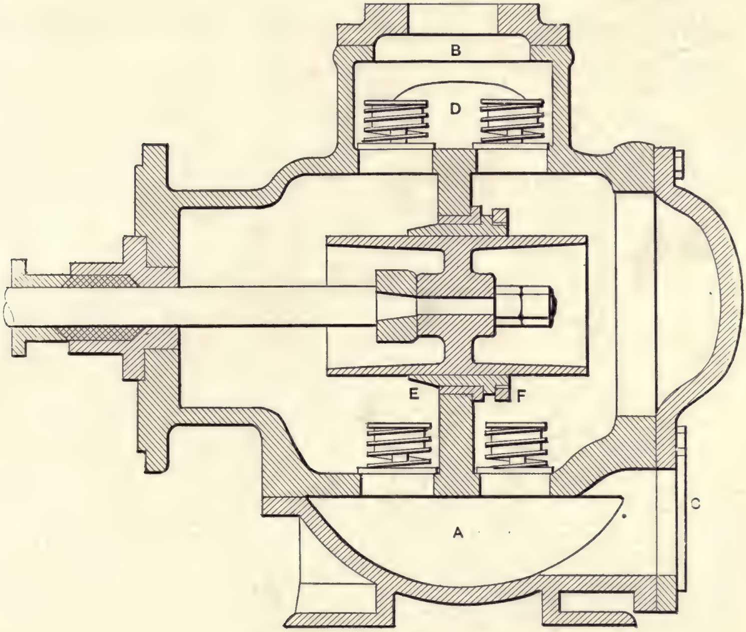


Fig. 217.—Plunger and Ring

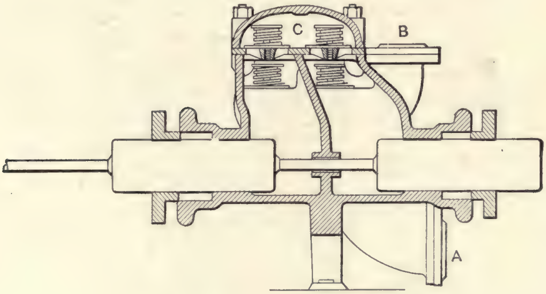


Fig. 218.—Worthington Pump.

pump is simple, and the water passages are all ample. The great disadvantage is in the possibility of leakage around the piston rod. To obviate this leakage the plungers are connected on the outside by a trombone frame (Fig. 219). The

rods E and F join the two cross heads G and H together. This pump may be the same in form as Fig. 218, with the exception of the connecting rod and sleeve.

A center outside-packed pump (Fig. 220) does away with

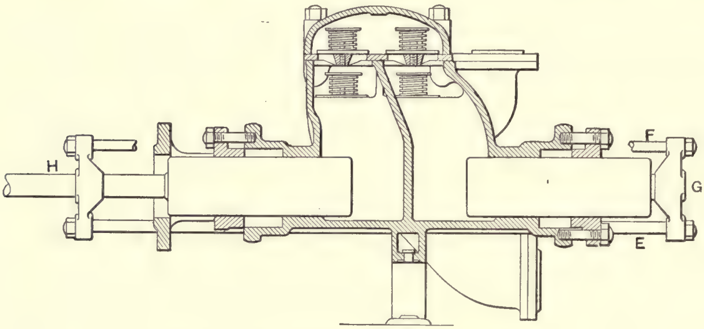


FIG. 219.—Outside-packed Plungers.

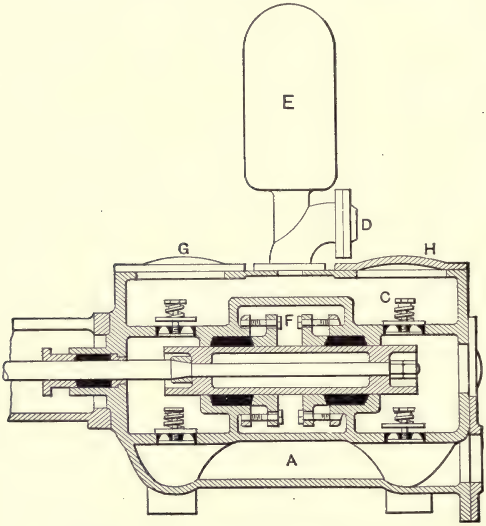


FIG. 220.—Center Outside Packing.

the necessity of outside rods and gives a pump in which all leakage past the plungers or rods is visible; the stuffing boxes are also clearly shown.

Water enters the suction chamber A from B, passes directly into the pump, and is driven into chamber C, and from there

into the discharge D. The space F is opened to the outside so that the stuffing boxes are always visible. As constructed in this figure the chambers C and A are cast solid with the other parts. This is not always the method of construction, as in many cases the two parts of the water end are separate castings connected by the chambers A and C.

The discharge valves are placed beneath hand plates G and H, while the suction valves may be examined and repaired through side hand holes in the pump barrel. The valves have removable seats and these are so placed that there is a direct

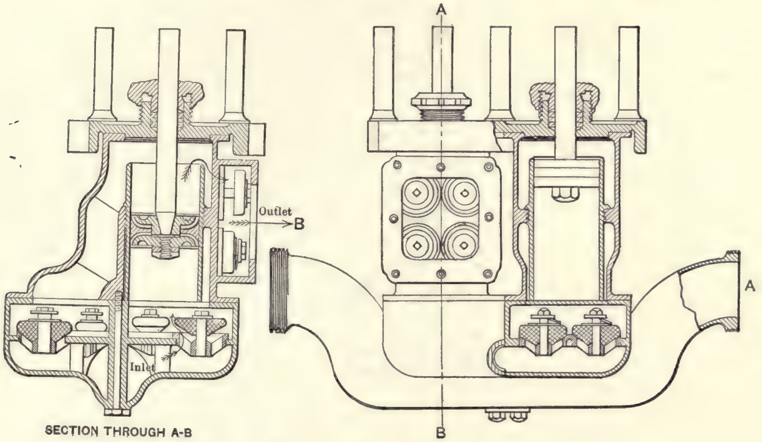


FIG. 221.—Fire Pump.

path of sufficient dimensions through the pump. Such arrangements cut down the friction and give a more smoothly acting pump.

These same points of a direct path, simplicity and ease of repair, are seen in the double-acting fire pump (Fig. 221), where the water enters at A and leaves at B. The piston is packed with cup leathers, and the passages from it to the suction valves are very large. The reason for this is the high rotative speed of these pumps and the necessity for having no interruption in the water column. There is not the need for such large passages on the discharge side as the water is being forced out. By removal of certain parts, these valves may be

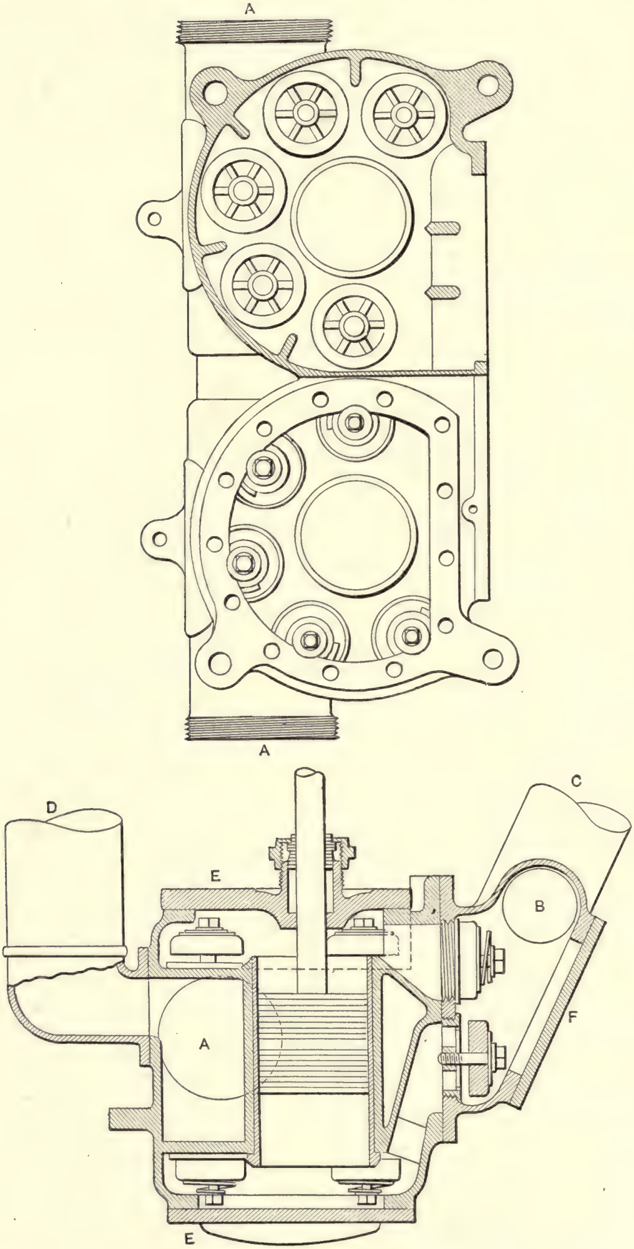


Fig. 222.—Metropolitan Fire Pump.

examined, although this is not so easily done as in most other pumps. These pumps are so closely built on account of the lack of room that many desirable features have to be sacrificed in order that more urgent needs may be met. The Ahrens

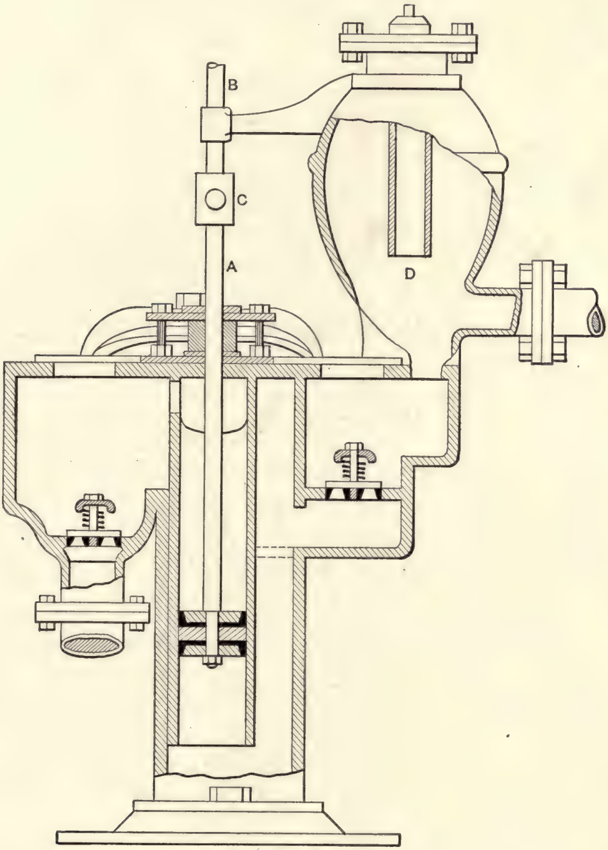


FIG. 223.—Railroad Pump.

pump has been improved by the American Fire Engine Co. in their Metropolitan engine. The suction valves (Fig. 222) have been replaced at each end of the cylinder through five suction valves of large area. The two discharge valves have not the same area as the suction, but friction here, although costly, will not interfere with the smooth running of the pump.

The discharge occurs through *B*. *C* is the discharge air chamber and *D* is the suction air chamber.

The valves of the pump are easily examined or replaced by the removal of the cylinder heads *EE* or hand-hole plate *F* over the discharge valves. When necessary the cylinder liner may be renewed. The piston is made deep and contains a number of grooves.

Fig. 223 is a detail of the water end of a pump used for railroad work. It is driven by a gas engine as shown in Fig. 147. There are four valves arranged around the pump barrel; two suction and two discharge. The suction valve shown in the figure is connected with the upper end of the cylinder, while the discharge valve is connected with the lower end. The valves not shown are connected with the opposite ends. The piston rod *A* is guided by the arm *B* while the cross-head *C* is connected with the pitman bar from a pin on a gear wheel. The air chamber *D* serves to steady the discharge.

The valve boxes are provided with covers which are held in place by yoke pieces so that the valves may be easily examined. The valve boxes are of ample size.

The barrels of deep-well pumps (Fig. 224) are usually made of a piece of brass tubing lowered into the well. The valves are of the disc-lift type or the ball type. The foot or suction valve is either lowered into place and held to its seat by its own weight or it may be put in place by attaching it to the pump rods and forcing it into position. It is necessary to have the foot valve water tight, as there will be lifting should the water be forced around it. These foot valves and buckets are usually packed with cup leathers.

The pump rods are often made of close-grain lumber with iron armored ends. One end is made into a nut, the other end is threaded as a bolt. In threading these ends considerable taper is used, so that it is only necessary to turn the rod three or four times to have ten or fifteen threads in contact when joining the sections. Precautions must be taken to prevent these screws from backing off, as the pump line depends on their holding.

Pressure pumps (Fig. 225) are made with heavy cylinder

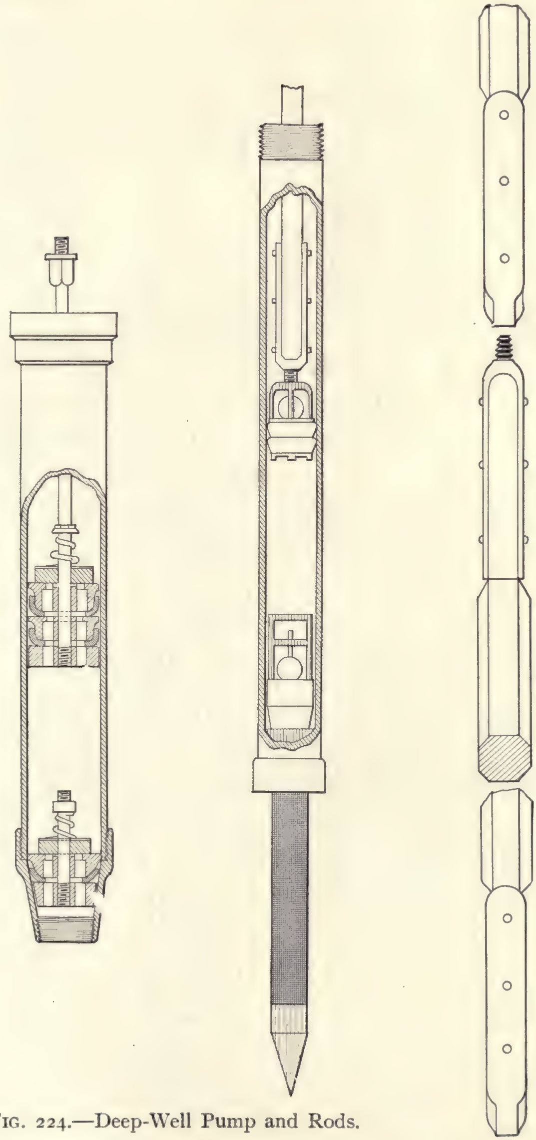


FIG. 224.—Deep-Well Pump and Rods.

walls, and in most cases they are of the plunger type, as this design is very suitable to high pressures. The packing may be

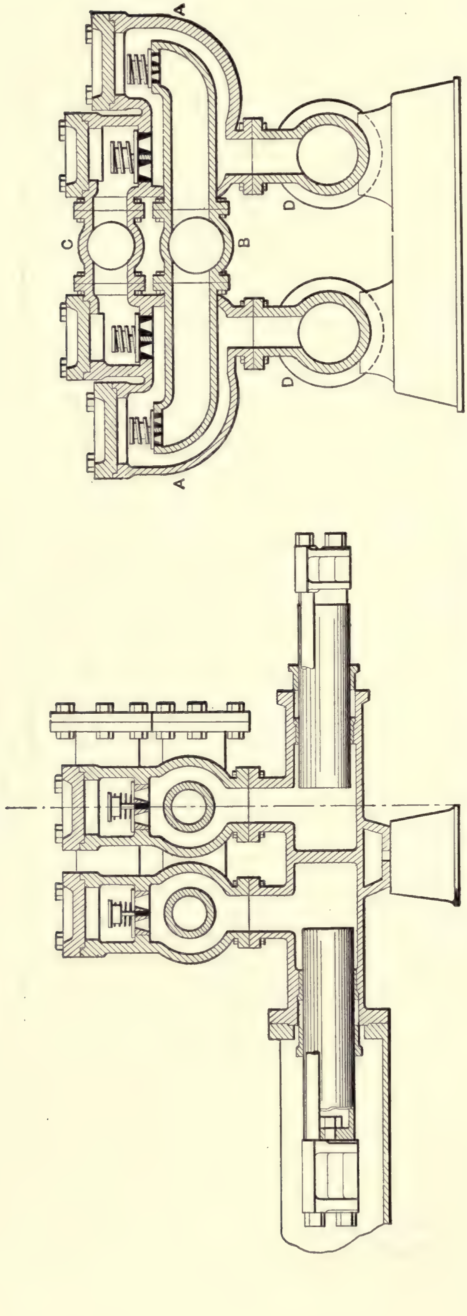


FIG. 225.—Pressure Pump.

of the ordinary hemp form shown in the figure or leather packing may be used. The valve chambers are usually single castings. They vary in form; sometimes the casting contains both suction and discharge valves, and at other times only one valve is in each casting. The valve castings *AA* are bolted to the suction casting *B*, and the discharge casting *C* as well as to the pump barrels *DD*. This arrangement gives a simple pump cylinder, and although the valve chambers are not simple, they are easily built, and machined. The valves are so placed that they may be examined and repaired by the removal of a cap. When these pressure pumps are made larger and more valves are required, the valve-box castings are increased in

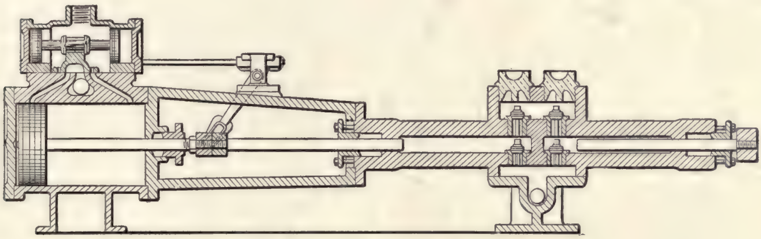


FIG. 226.—Pressure Pump.

In smaller pumps under very high pressure the valve chambers may be cast with the pump barrel as shown in Fig. 226, which represents a high-pressure Burnham pump. The small valves are so arranged that by removing one plug both suction and discharge valves may be examined. Attention is called to the great difference between the plunger and the steam piston areas and to the thickness of the cylinder wall. The valve seats are renewable, so that if any of these should wear it could easily be replaced without disconnecting the pump.

The water end of large pumps has been greatly simplified. One of the older pumps for the Boston sewage system, designed by Mr. E. D. Leavitt, Jr., is shown in Fig. 227. In this case the valves were rectangular clack valves, $3\frac{3}{4} \times 13\frac{1}{2}$ inches. Six

of them were attached to a frame and placed on the openings of the suction, while three discharge valves were attached to each frame on the other side. The frame with the suction valves open as would occur on the suction stroke is shown on the left, while on the right the frame alone is to be seen in section and at the upper right hand the discharge valves are shown closed.

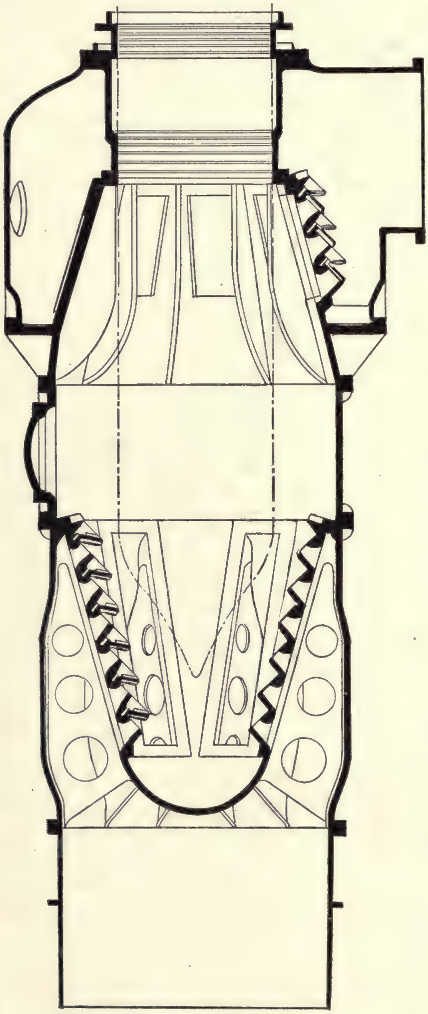


FIG. 227.—Sewage Pump.

The suction valves and discharge valves are in the proportion of 36 to 27. The reason for this was seen in Chapter V. In all cases an endeavor is made to cut down the losses on the suction side. The clear passage to and from the valves is necessary, and valves of large areas with no supports obstructing the passage are required to pass the solid matter which is found in sewage. Mr. Leavitt states that there is a record of this pump having passed through its water cylinder a plank $2 \times 12 \times 36$ inches.

The water end is 48 inches in diameter and of 108 inch stroke. The form of the plunger is dotted in the lowest position. The form of stuffing box is illustrated together with the grooves on the inside of the gland and bushing. This form of so-called

labyrinth packing has been proven of little value. The manhole in the center ring of the pump permits entrance

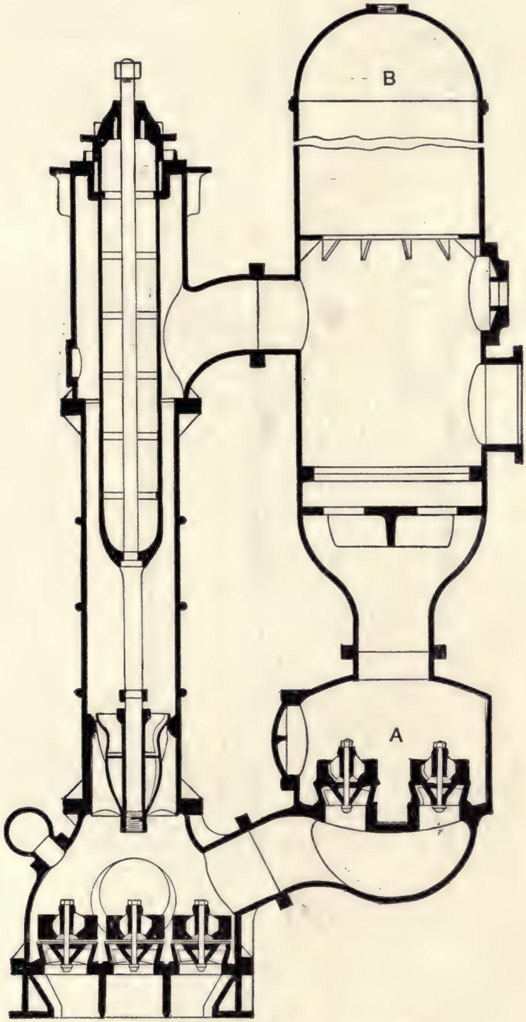


FIG. 228.—Lawrence Pump.

for the examination of the suction valves, while a manhole in the discharge chamber permits the examination of the other valves. The cylindrical casting is made in three principal

parts to simplify the foundry work and to make shipping and erection less difficult.

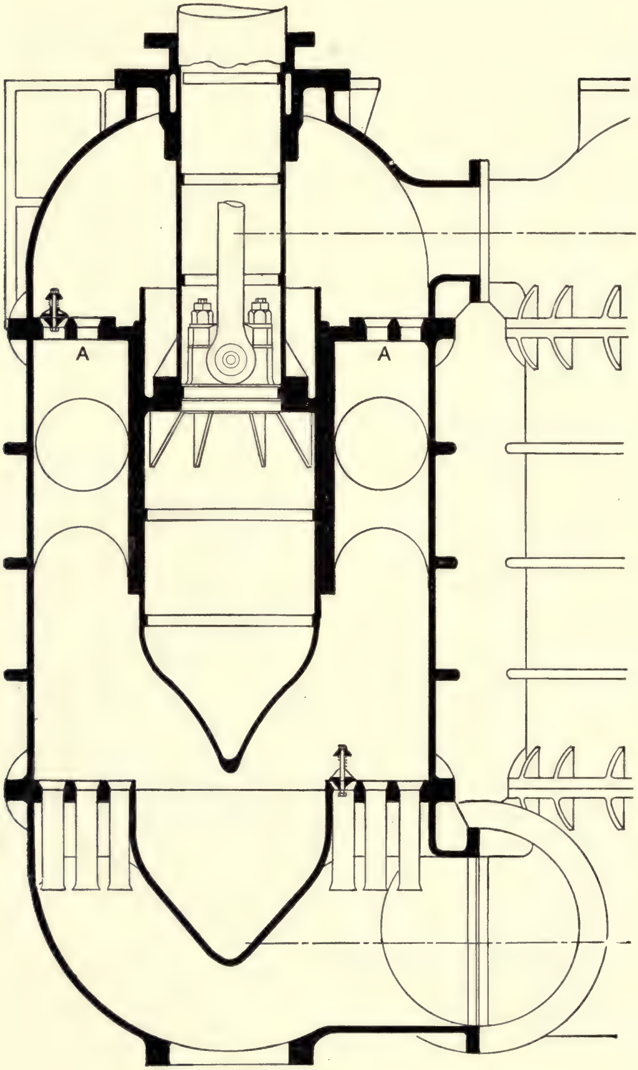


FIG. 229.—Ontario Pump.

Fig. 228 illustrates the Lawrence Water Works pump of Mr. E. D. Leavitt, Jr. It is the bucket form of pump with a

supplementary discharge at *A* for the purpose of giving a passage to the water should the valve in the bucket cease to operate. When the pump operates as a bucket pump the course of the water is direct through the pump barrel. The use of a plunger enlargement on the rod of one-half the area of the piston serves to produce a discharge on each stroke, while suction occurs on every other stroke.

The valves for this pump were originally 16-inch brass double-beat valves shown in Fig. 252, but as the friction of the

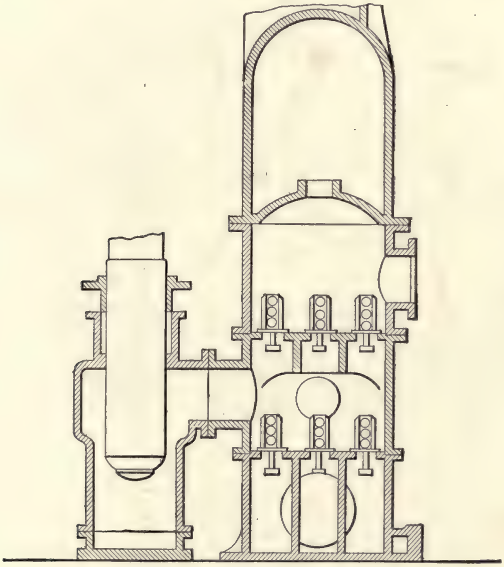


FIG. 230.—Milwaukee Pump.

pump was excessive the valves were changed to the annular form as shown in Figs. 228 and 250. This reduced the friction very materially. The castings making up the pump, the air chamber *B*, the manholes, valve decks and other details are seen by a study of the picture.

A single-suction double-discharge plunger pump is shown in Fig. 229. In this type of pump for Ontario, Leavitt used a large number of small valves and on the suction side small draft tubes were placed below the valves. The plunger was

shaped to fit into the bottom of the cylinder and cause all of the water to be in circulation. Such a construction may be questioned unless there is a lack of head room and it is thought necessary to point the plunger. The plunger has an extension sleeve on it of one-half the area of the plunger. The castings are simple and well braced by brackets and webs, as well as

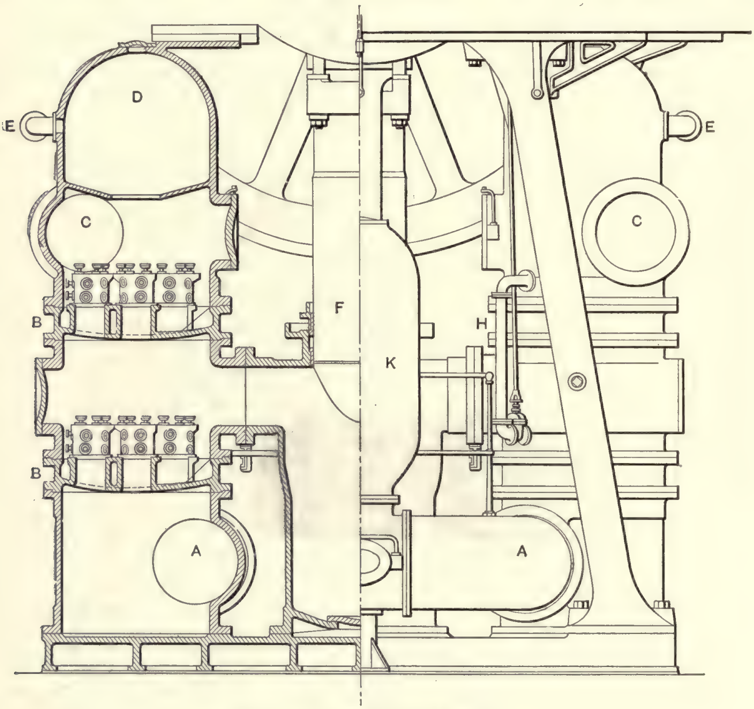


FIG. 231.—Cincinnati Pump.

reinforced by rings around the barrel. The path of the water is direct, and the head pressure on the discharge valve deck is supported by the webs *AA*, which act as girders.

The valves are spring controlled, of small size, and shown in greater detail in Fig. 252.

The Allis pump of Milwaukee (Fig. 230) is another case in which a series of lift valves were used to replace large double-beat valves. In this case, each large valve was replaced by a box or cage on the sides and top of which were openings for

small valves as shown in detail in Fig. 254. The construction shown in Fig. 230 illustrates how simple the castings of a large pump may be. The valve decks, which are reinforced by cross ribs or girders to withstand the pressure, are the end parts of individual castings; on account of this the faces may be easily machined in the shop, and erected in the field. The path of

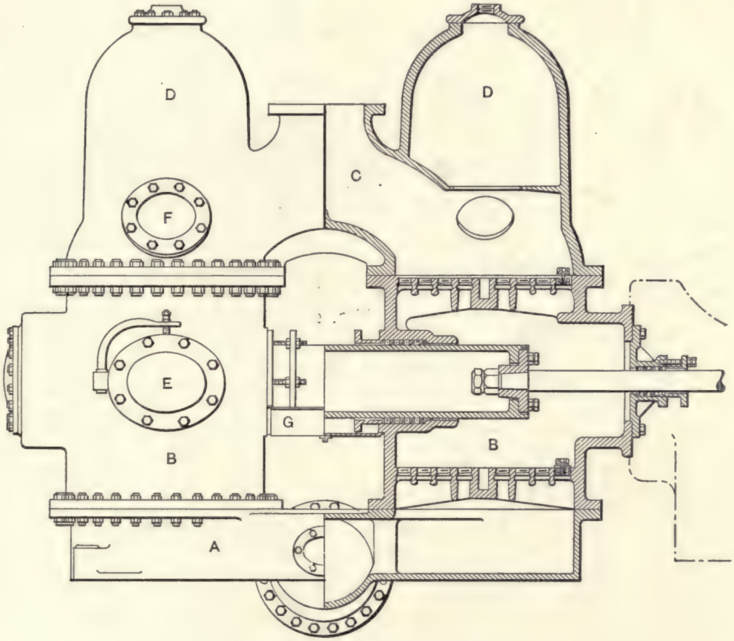


FIG. 232.—Snow Pump End.

the water is more or less direct and the suction pipe is of proper size for this slow-running pump.

Fig. 231 illustrates a 30,000,000-gallon water end built by the Holly Pump Co. The suction pipe enters the valve chamber on each side of the pump at *AA*; the valve decks *BB* are made of similar castings of considerable depth to withstand the high working pressure. The valves are mounted on cages as in the previous figure. The discharge occurs at *CC*. The space *D* in the top of the discharge acts as an air chamber. The pipes *EE* are used to introduce compressed air into this space. The

plunger *F* draws water from the suction valves on each side of it on its up stroke and discharges through the two sets of discharge valves on its down stroke. The connection *H* made between the discharge side and pump space is used to prime the suction valves when necessary, but may also be used in starting the pump to equalize the pressure on the plunger on each stroke. A large suction air chamber is shown at *K*.

Fig. 232 shows the water end of a large horizontal pump.

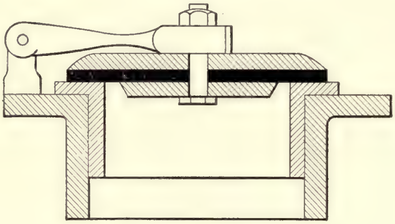


Fig. 234.—Metal Hinge Clack Valve.

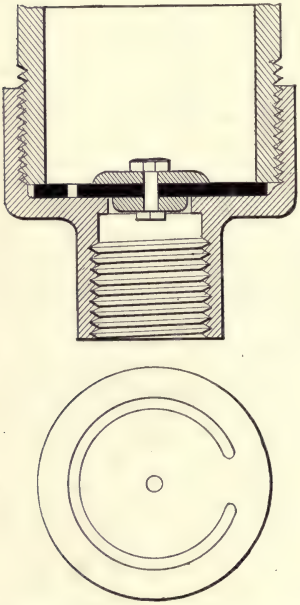


Fig. 233.—Leather Clack Valve.

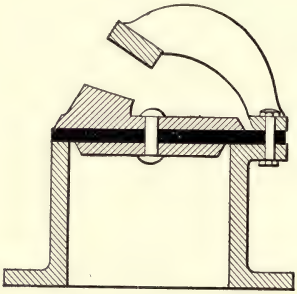


Fig. 235.—Clack Valve.

The water end is made up of four principal castings: A suction chamber *A*; two pump ends *BB*; and a discharge chamber *C*, containing the air chambers *DD*. The valve decks in the pump ends *BB* contain a large number of small valves; they are well braced by cross ribs as seen in the figure. The manhole cover at *E*, which is hung from an arm, serves for the examination of the suction valves, while that at *F* is used for the discharge. The figure illustrates very clearly the method of packing the plunger and piston rods and the simple manner in

which the castings can be made. The trough G beneath the intermediate stuffing boxes is intended to catch the drip.

One of the simplest valves found in small hand pumps is the leather clack valve (Fig. 233). A circular piece of leather shown in plan in the figure has a groove cut out of it and after fastening two iron washers to this, the leather is held beneath the pump barrel and the valve seat casting. The small part of the leather left after cutting the groove forms a hinge. The upper washer is not only used to weight the clack, but it also supports the weight of water above the leather, making a water-tight valve.

In Fig. 234 the leather hinge has been replaced by a pin, and in Fig. 235 the leather hinge is retained for a valve which is

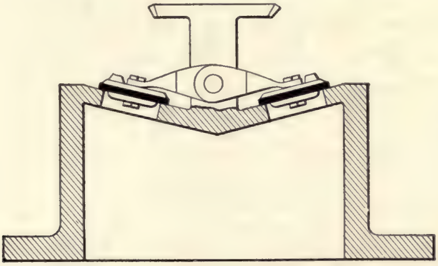


FIG. 236.—Butterfly Valve.

rectangular. Such a valve is used for sewage pumping (Fig. 227). A double clack valve (Fig. 236) is sometimes called a butterfly valve. In this form the valve is made rectangular. Such a shape is often found when clack valves are used. The valve seat is formed separate from the pump casting and is bolted in position. Stops are usually employed to keep the valve from opening too much.

A metal clack valve (Fig. 237) may be used at times where the liquid handled would destroy the leather. The pivot in this case moves in a slot to allow for the wear of the valve. A pin through a hole would not allow the valve to seat properly after wear had occurred.

The conical valve of Fig. 238 is guided by wings on its lower face. The seat of this valve is often made of bronze, and

inserted in the valve deck. This renders it an easy matter to renew the seat and permit one to use a better metal for it, the valve deck being made of a soft iron. The valve is fitted to the seat by grinding. The operation consists in turning the valve against the seat by a screwdriver, after oil and emery are introduced between them. To give the valve in seating

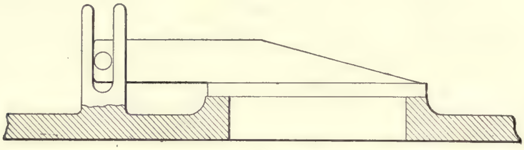


FIG. 237.—Metal Clack Valve.

a turning motion, the wings below the valve disc are bent into a helical form (Fig. 239). The action of the water on the vanes is to rotate the disc. This allows the valve to seat at different points each time, thus eliminating the excessive grooving which occurs when a valve always seats at the same point after the formation of an incipient groove. The rotary motion also gives the valve a wiping action in seating.

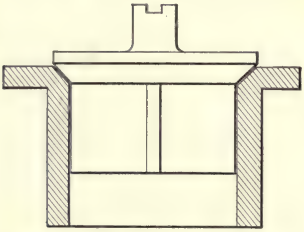


FIG. 238.—Conical Valve.

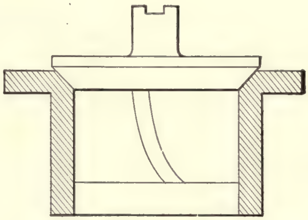


FIG. 239.—Helical Wings.

The ball valve (Fig. 240) is an effective type. The action of the cage over the ball and the removable nature of all parts are evident from the figure.

One of the common forms of valves used with all forms of pumps is shown in Fig. 241. The valve seat is made of composition brass, and screws into the valve deck. The valve is made of a composition of rubber and other substances. It is backed by a piece of sheet brass against which the spring presses.

The spring is held beneath a nut on the valve spindle. The spindle is screwed down tight against a shoulder so that it

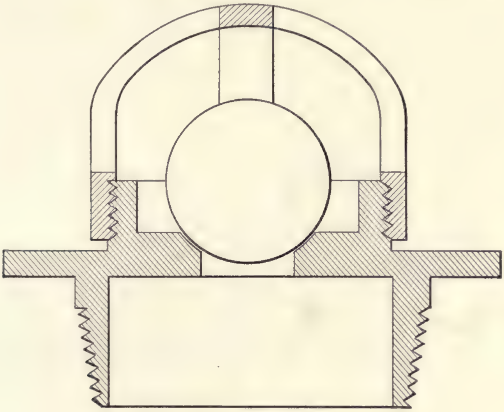


FIG. 240.—Ball Valve.

will not back off. A split pin put in the hole in the top of the spindle prevents the nut from unscrewing. All parts shown

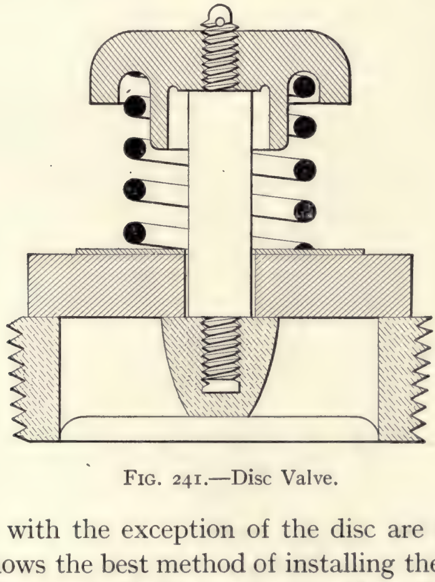


FIG. 241.—Disc Valve.

in the figure with the exception of the disc are made of brass. The figure shows the best method of installing the valve spindle, as the spindle remains in the seat when the valve is removed.

At times, however, the spindle and nut are combined into one piece, Fig. 242, and in placing this in position a plug wrench is used.

The valve disc is sometimes backed up by a brass cup

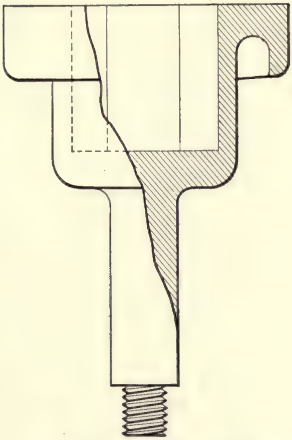


Fig. 242.—Valve Spindle.

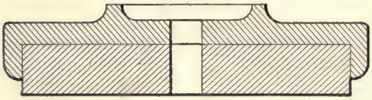


Fig. 243.—Valve Backing.

Fig. 244.—Metal Disc Valve.

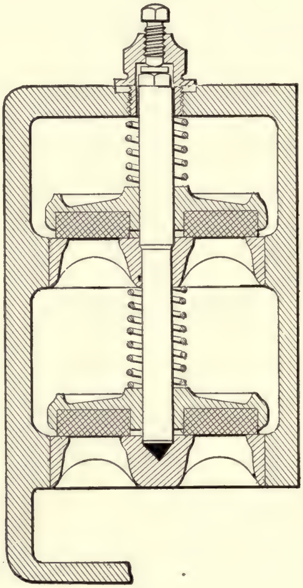


Fig. 245.—Cameron Valves.

into which it fits (Fig. 243). This gives more stiffness to the valve.

When hot water is used brass valves replace the composition valves which are intended for cold water, although there are special compositions used for hot water. Fig. 244 illustrates the form often used for metal valves.

A neat arrangement for the suction and discharge valves is made by the Cameron Company. These valves (Fig. 245)

are placed on a common spindle which is held in place by a set screw in a cap, screwed into the wall of the valve chest. This spindle also serves to hold the valve seats in place. These valves are backed with metal cups.

The Marsh pump is equipped with metal valves the spindles

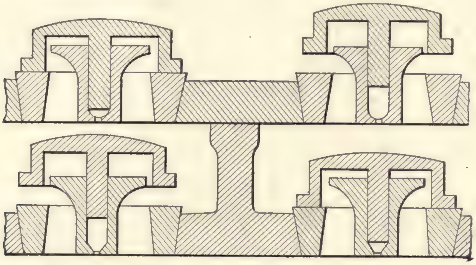


FIG. 246.—Marsh Valves.

of which are guided by holes in the metal at the center of the valve seat. This center is so formed that the water is deflected under the valve as shown in Fig. 246. This central part is almost as large as the cavity of the valve and when the valves are raised the distance shown in the figure, there is no tendency

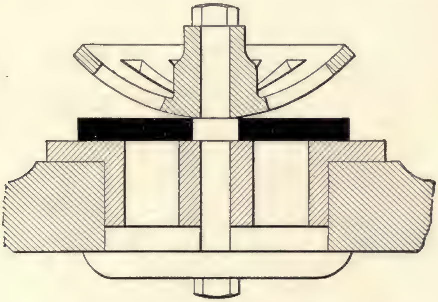


FIG. 247.—Rubber Valve with Guard.

for them to lift higher. This puts them into a position where they may seat quickly, but without shock, as the deflector acts as the piston of a dash pot.

When rubber valves are made of large diameter they are designed as shown in Fig. 247. The guard on the back of the valve keeps the valve from opening too far. The holes in the

center permit the water to act on the back of the disc and aid in quick closing. The method of supporting the spindle, and thus holding the seat in position, is clear from the figure.

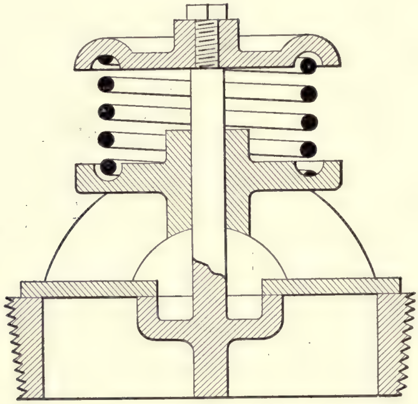


FIG. 248.—Double Ported Valve.

To increase the opening in the valve discs of large diameter they are made multiported; in reality they become a series of concentric rings, joined by radial arms, as was explained in

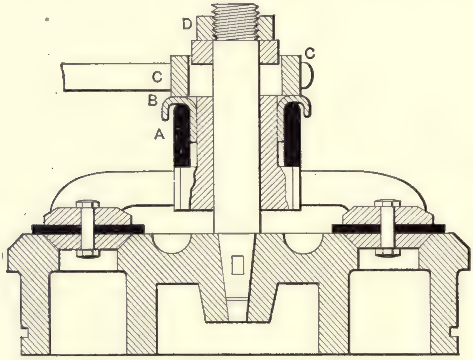


FIG. 249.—Riedler Valve.

Chapter V. Fig. 248 illustrates such a valve with seat, spindle, spring, and nut where one ring only is used. The form of this type of valve, used by the Allis-Chalmers Co. in their Riedler pumps, is best shown by Fig. 249. In this valve a leather

washer is used in addition to the conical-faced ring and seat. The leather is held between two iron rings. This leather really makes the joint when the pump is in action. The valve may be raised from its seat by the water pressure when the arms *CC* are raised. The amount of motion is limited by the nut *D* on the spindle. At the end of the stroke the arms *CC* are driven down against the metal sleeve *B* by the action of an eccentric. This presses against a rubber collar *A* and forces the valve to its seat. The rubber collar gives a yielding

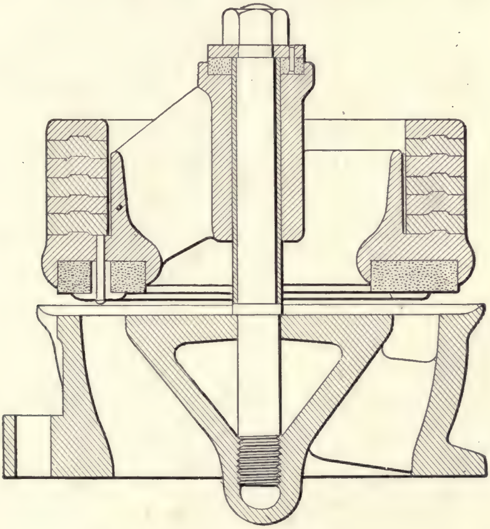


FIG. 250.—Weighted Valve.

connection to care for the possibility of solid objects getting beneath the valve. Before the pump reaches the other end of its stroke the arms *CC* are raised so that the valve may open as soon as the pressure is sufficient. The Riedler valve was first employed with express pumps.

A weighted ring valve of American design is shown in Fig. 250, although for smaller sizes springs could be used, as shown in Fig. 251. This valve (Fig. 250) was employed on the Lawrence pump to replace the double-beat valve of Fig. 252. Where large area is required, double-beat valves (Figs. 252

and 253) are used. Double-beat valves are valves containing two seats, as at A and B. The upper seat B is made smaller than that at A, and hence an upward pressure from below causes the valve to lift from its seat and the water to escape by these two openings. This gives a large area and by properly arranging the size of each seat, a proper amount of pressure increase can be had to lift the valve weight. The motion of the valve is limited by the nut on the spindle. In Fig. 252 the valve is open to its full extent. Although these valves have been used to some degree, the American practice is to use a number of small valves, and where sufficient valve deck area is not available valve caps (Fig. 254) may be placed over the openings used for double-beat valves. In this manner a large discharge area may be obtained. The valve cap or box is held to the main valve deck by a large through bolt. A series of webs are employed within this to stiffen it.

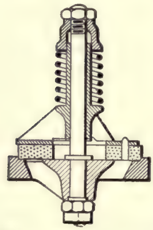


FIG. 251.—Double Ported Valve.

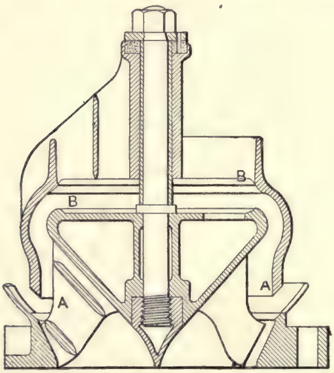


FIG. 252.—Double Beat Valve.

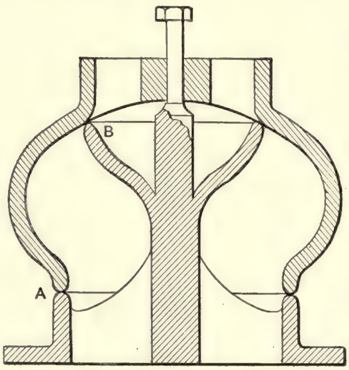


FIG. 253.—Double Beat Valve.

The valves shown in Fig. 255 illustrate the method used for high-speed pumps. These valves have been recently introduced by Witting of England. In them, two brass rings are placed around each opening. The rubber rings BB force the brass rings together when the water pressure on each side of the brass ring is the same. The rings touch each other, and are

held together by the water pressure on the outside as well as by the rubber. When the internal pressure is sufficient to force

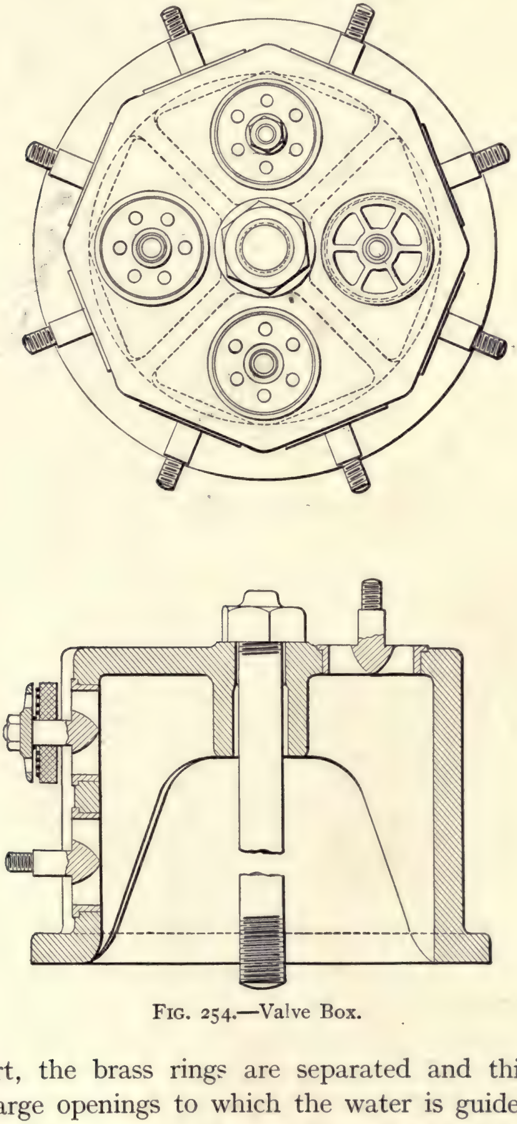


FIG. 254.—Valve Box.

these apart, the brass rings are separated and this gives a series of large openings to which the water is guided. These valves may be made with leather facings, and helical springs may replace the rubber. The lower figure illustrates a method

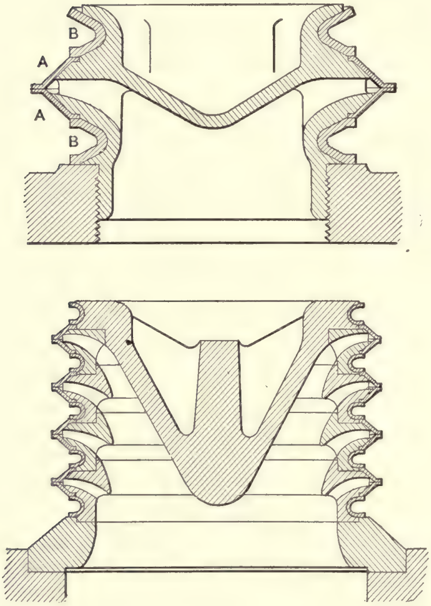


FIG. 255.—Witting's Metallic Valves.

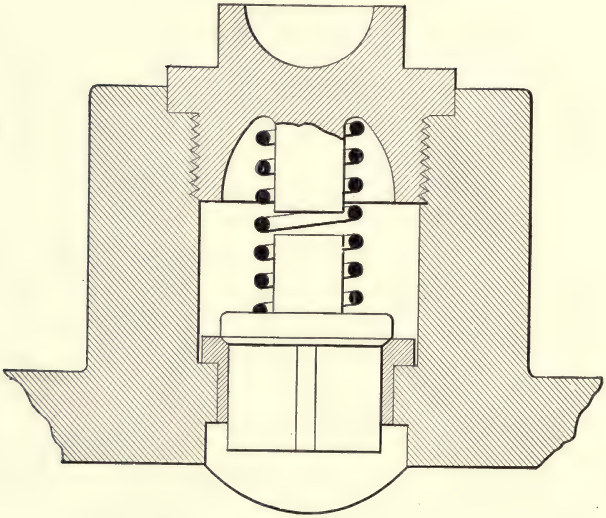


FIG. 256.—Pressure Valve.

of increasing the valve area. These valves are so constructed that they may replace other valves, the cages being bolted to the valve decks. These valves are somewhat similar to the

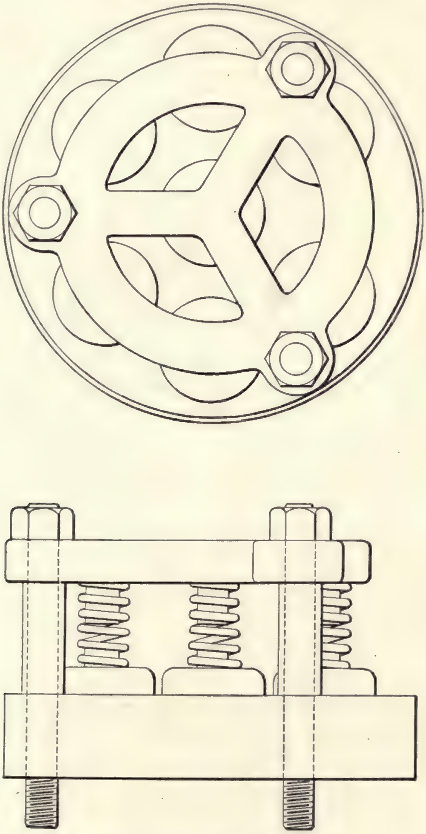


FIG. 257.—Pressure Valves.

valves used on Merryweather's fire engine of 1870, in which rubber replaced the metal rings.

In high-pressure pumps the valves are made deep for stiffness; the springs are usually made heavier on the discharge side to close the valve quickly, as considerable spring pressure is not objectionable on this side. The cap over the valve and the walls are made heavy and the valve should be accessible. Fig. 256 illustrates a simple form of valve, as used with

the Burnham pump, while Fig. 257 shows a series of valves on one deck. The springs are held in place by a ring. This makes it possible to remove the cover of the valve chamber for inspection without disturbing the springs. In these two arrangements the springs are guided by spindles on the caps, frame, or valve. When this cannot be done a frame is formed on top of the valve and placed around the valve and its frame (Fig. 258). Such a cage may be held down by a ring and bolts (Fig. 259), as was also used in Fig. 257. The massive construction shown in the last four figures is due to the excessive

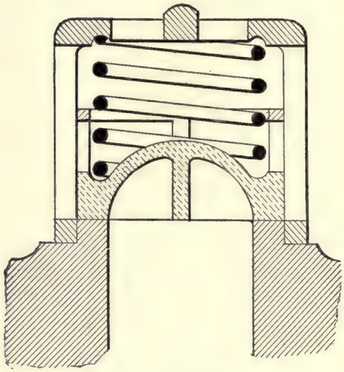


FIG. 258.—Pressure Valve and Cage.

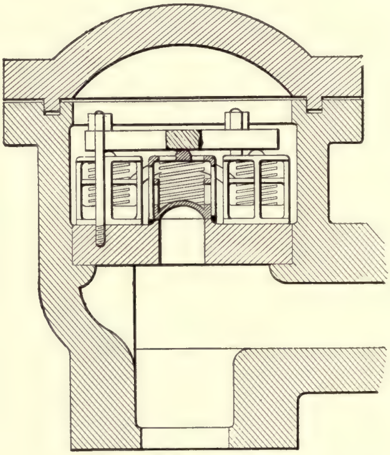


FIG. 259.—Valve Pot.

pressure carried. In all of them, it is seen that the parts may be easily removed and repaired, and quick seating of the valve is possible so that the slip may be cut to a minimum.

A form of valve used by German pump builders is known as the Gutermuth valve.

In this valve (Fig. 260) a sheet of steel or bronze is bent into the form of a spiral which forms a spring for the end of the strip, which is left flat to form the valve. A pressure from beneath drives the plate into the dotted position, and the water has a path through which the flow occurs almost straight from the channel leading to the valve. This channel is not arranged

normal to the valve disc, as it is desired to continue the flow in as straight a line as possible.

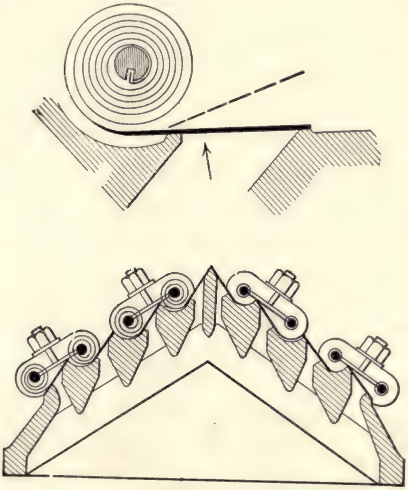


FIG. 260.—Gutermuth Valve.

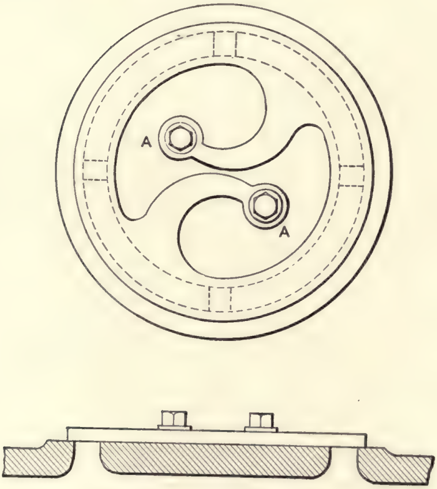


FIG. 261.—Borsig Valve.

The Borsig Co., of Germany, use a metal valve of very simple form. It is made of thin spring metal of the form shown in Fig. 261. The points *AA* are bolted to the valve deck,

and the arms extending to the outer ring form the springs to seat the valve. At times additional helical springs are placed on the back of this.

The discharge air chambers for pumps are made in various

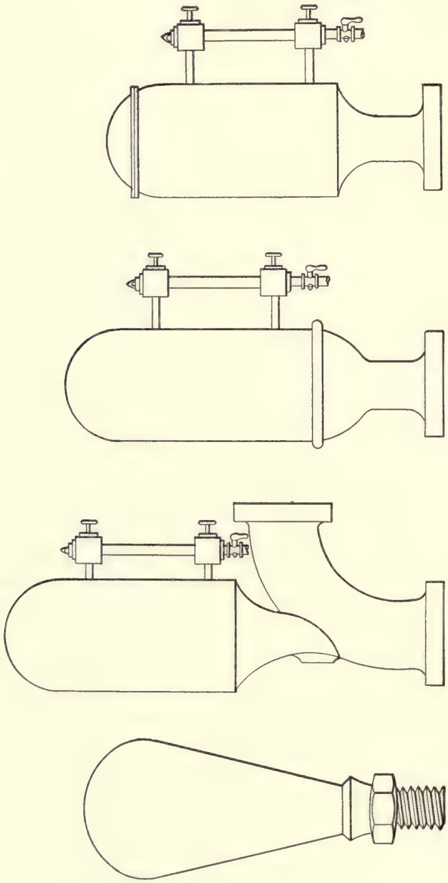


Fig. 262.—Air Chambers.

forms, as shown in Fig. 262. For small pumps the first form is often used with the main body of sheet copper, shaped into a conical vessel. The other forms are usually made of cast iron, and are placed on top of the discharge valve chamber. The second form is well arranged, as the main discharge is cared for by the bend in the pipe. The sizes of these are fixed by

considerations given in Chapter V, and the thickness is determined by the formulæ given in this chapter. As was pointed out earlier, the size of the air chamber will depend on the details of an installation, so that a standard size cannot be fixed for stock pumps which will be suitable for any purpose.

The air from the chamber is gradually absorbed by the water, and to keep the chambers charged air compressors may be used. A simple device (Fig. 263) is sometimes applied. This consists of piping attached to the pump cylinder and air chamber in which are placed two check valves and one gate valve. The check valves open toward the air chamber. When the valve is opened air is drawn into the vertical line on the suction stroke of the pump, and on the discharge stroke this is compressed and sent over into the air chamber, as the air cannot escape to the atmosphere and the pressure in the cylinder is greater than the pressure in the air chamber. On each stroke there is a slight discharge of air into the chamber, and after sufficient air is obtained the stop valve is shut off and the operation of this air pump ceases.

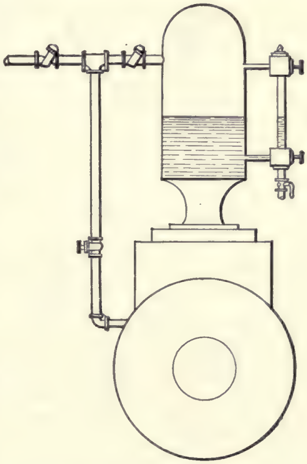


FIG. 263.—Air Chamber Pump.

The air chambers on the suction side of the pump are important adjuncts, as was seen in Chapter V. They should be placed so that they can receive and care for the water which it is necessary to store for the variation in the suction of the pump. The air chambers of Fig. 264 are all placed at the end of a suction line, while the water is taken to the valve chest from the side of this line. The various arrangements illustrate how the air chamber can be placed when different conditions of piping exist. In all cases the water is flowing directly into the air chamber.

Fig. 265 shows two air chambers; the left-hand being a simple one, made of pipe fittings, and of proper design, while

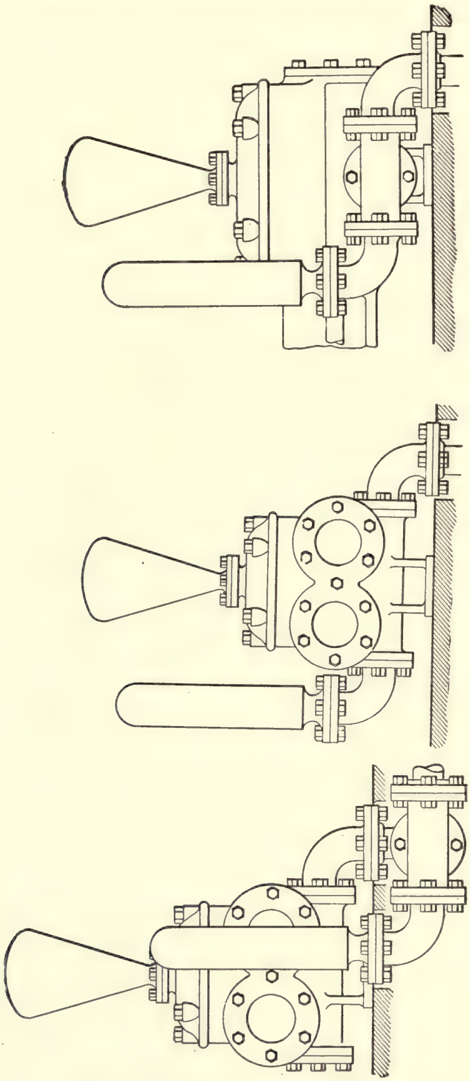


Fig. 264.—Arrangement of Suction Piping.

the right is improperly arranged, since the water, in passing at right angles to the air chamber, must exert some force on the valves beyond before the air in the chamber is compressed.

The left-hand chamber discharges from the end B at a varying rate, while the supply at A is uniform. The excess or deficiency is made up from the water which passes into the space around the upper pipe. This flow goes naturally into the space, as the stream enlarges suddenly. To cut down the loss due to sudden enlargement and yet get the advantage of it, the suction air chamber could be built as shown in Fig. 266, and in this manner

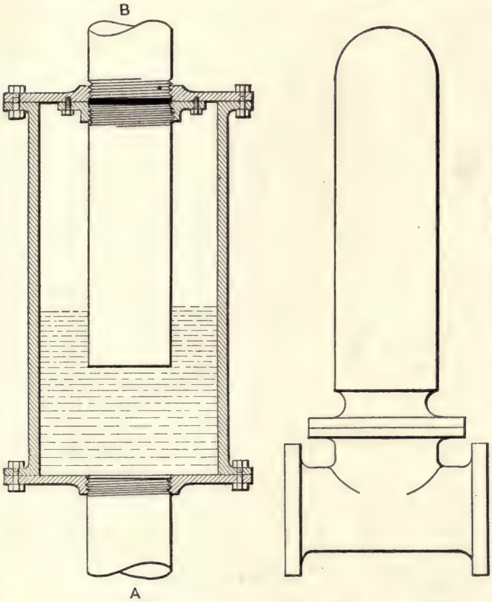


FIG. 265.—Suction Air Chambers.

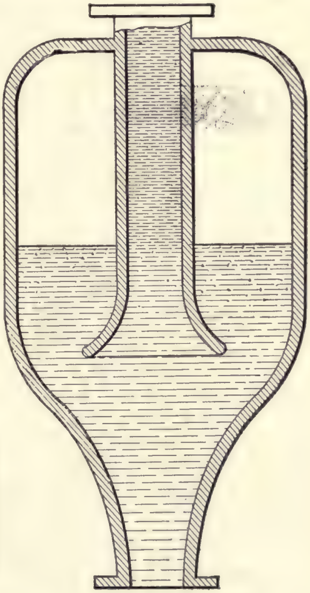


FIG. 266.—Suction Air Chamber.

the losses of sudden enlargement and contraction could be reduced.

To protect the pump valves against leaves, twigs, and other foreign particles which might lodge beneath the valves and prevent their proper action, strainers (Figs. 267, 268, 269 and 270) are used. These are made of wires, grills, or drilled plates. They should always be so constructed that they may be cleaned easily. At times the strainer is combined with a foot valve, as in Figs. 269 and 270. A foot valve is a large valve or series of small valves attached to a valve deck as shown in the figures, and placed on the end of the suction main. The purpose of

the foot valve is to hold the water in the suction main when the pump is shut down, and to keep the line filled in case it is very long. The large foot valve may be of the lift form, as shown

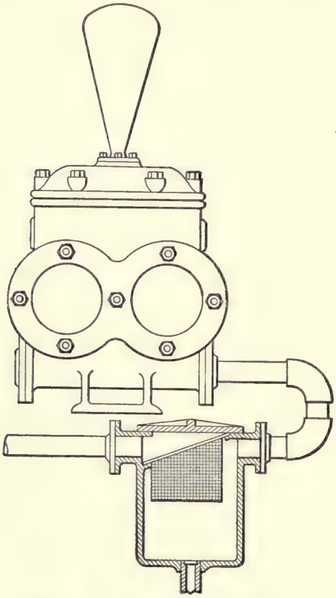


FIG. 267.—Suction Strainer.

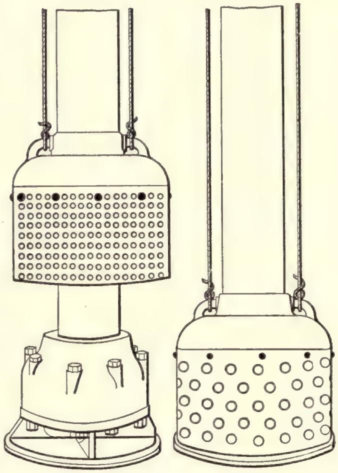


FIG. 269.—Foot Valve and Strainer.

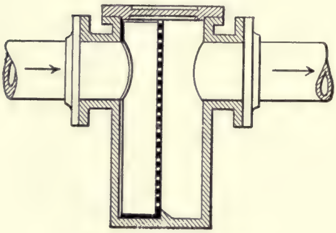


FIG. 268.—Strainer.

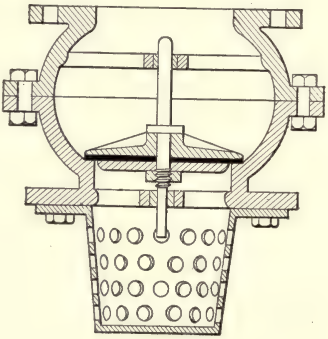


FIG. 270.—Foot Valve and Strainer.

in Fig. 270, or it may be of the form shown in Fig. 271. The type shown in Fig. 268 is arranged as shown in Fig. 271.

The suction pipe should be of large diameter, as straight and as short as possible, and air tight. The smallest leak is apt to destroy the proper action of the pump. It is well to test these pipes under pressure before covering them. The

pipe must be laid on a uniform grade to prevent air pockets. Everything should be done to cut down the losses, as the total loss with the lift must be kept less than 34 feet.

With these details in mind, the design will now be continued. The number of revolutions, the size of the pump, the size and number of the valves, the sizes of the pipe lines,

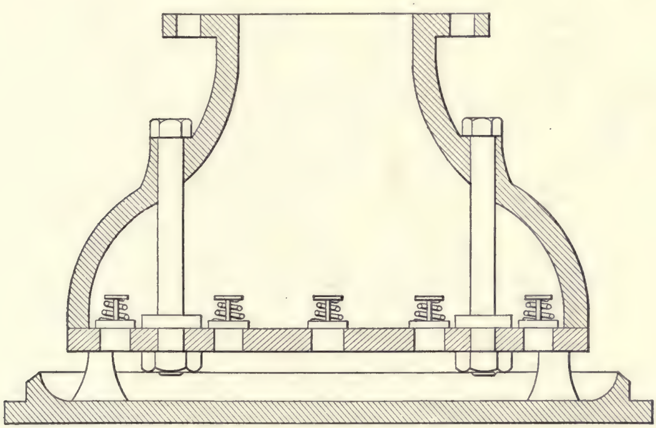


FIG. 271.—Foot Valves.

and the size of the air chambers are determined as in Chapter V. From these, the pressure in the water cylinder can be found.

DESIGN OF CYLINDER

In the water cylinder (Fig. 272) the first quantity to be determined is the thickness of the walls.

When the cylinder walls are cylindrical and the surface is not subject to wear, the formula:

$pd = 2tS$ or its equivalent may be used;
p = the water pressure in lbs. per sq.in.;
d = diameter to center of walls in inches;
t = thickness in inches;
S = allowable stress in lbs. per sq.in.

This formula is for thin cylinders; for thick cylinders used

with heavy pressures the formulæ of Barlow and Lamé are employed.

Before proceeding with these formulæ it will be well to note the table (page 309) of the strengths and other properties of the materials to be used in design together with the safety factors required. These values have been taken from standard

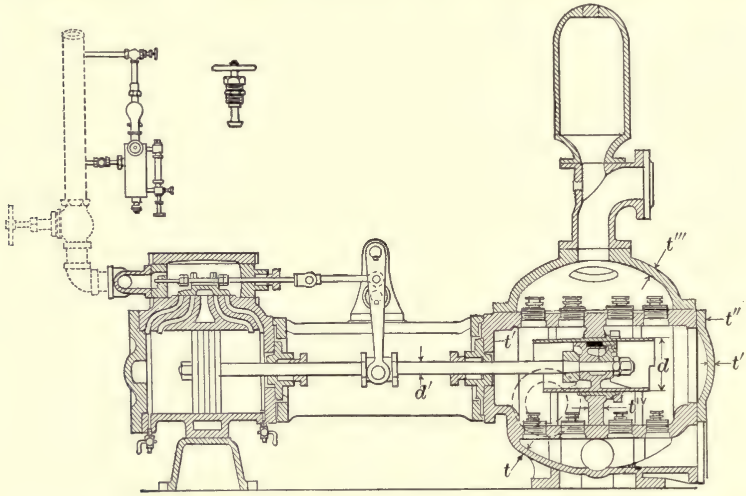


FIG. 272.—Pump Section.

texts on machine design as well as from engineering handbooks and current literature.

The formula above where cast iron is used as the cylinder wall is put into the form

$$t=\frac{pd}{5000}+0.25''.$$

When the cylinder is bored for the reception of the piston and allowance is made for reboring, this equation becomes

$$t=\frac{pd}{5000}+0.6''.$$

In making the head as shown in the Fig. 272, the radius of the spherical portion should be made equal to the diameter

TABLE OF PROPERTIES OF MATERIALS

Material.	Tension.		Compression.		Shear.		Elongation.	Modulus of Elasticity.	Factors of Safety.			Weight in Pounds.	
	Elastic Limit.	Ultimate. St'ngth.	Elastic Limit.	Ultimate St'ngth.	Elastic Limit.	Ultimate St'ngth.			Steady Load.	Live Load.	Shock.	Per Cubic Foot.	Per Cubic Inch.
Cast iron	—	20,000	—	100,000	—	10,000	—	20,000,000	6	12	18	450	.26
Wrought iron	28,000	55,000	30,000	55,000	22,000	44,000	15% in 8″	29,000,000	4	6	12	480	.278
Malleable iron	—	35,000	—	—	—	—	—	—	4	6	12	—	—
Mild steel	37,500	65,000	—	65,000	—	55,000	25% in 2″	30,000,000	4	6	12	490	.284
Steel forged	37,000	80,000	—	90,000	—	64,000	23% in 2″	30,000,000	4	6	12	490	.284
Steel-forged annealed nickel	45,000	80,000	—	—	—	—	23%	—	4	6	12	—	—
Steel-forged oil-tempered nickel	60,000	90,000	—	—	—	—	22%	—	4	6	12	—	—
Steel castings	30,000	60,000	—	—	—	—	20% in 2″	—	4	6	12	—	—
Steel, 0.96% carbon	69,000	118,000	—	—	—	—	—	30,000,000	4	6	12	490	.284
Copper cast	—	20,000	—	—	—	—	—	15,000,000	6	12	—	540	.314
Copper rolled	5,600	35,000	4,000	—	3,000	—	—	—	6	12	—	—	—
Bronze	6,200	30,000	—	—	4,000	—	—	13,500,000	10	15	—	540	.314
Phosphor bronze	19,700	58,000	—	—	14,500	43,000	—	14,000,000	5	10	15	—	—
Tobin bronze, cold rolled	—	100,000	—	—	—	—	—	—	5	10	15	—	—
Tobin bronze, rolled	—	75,000	—	—	—	—	15% in 2″	—	5	10	—	504	.291
Muntz metal	—	49,000	—	—	—	—	—	—	5	10	—	512	.296
Manganese bronze	30,000	63,000	—	120,000	—	—	10%	—	5	10	15	—	—
Aluminum castings	6,500	15,000	—	12,000	—	12,000	—	11,000	5	10	15	166	.096
Pine	—	12,000	—	6,000	3,000		—	1,600,000	6	10	15	40	—
Oak	—	15,000	—	8,000	4,000		—	1,700,000	6	10	15	50	—
Ash	—	15,000	—	8,000	4,000		—	1,600,000	6	10	15	45	—
Brick masonry	—	—	—	1,400	—		—	—	6	10	—	125	—
Stone masonry	—	—	—	240 allowable		—	—	—	6	10	—	140	—
Concrete	—	—	—	150 "		—	—	—	10	15	—	140	—
Leather	—	4,200	—	—	—		—	25,000	—	10	—	—	—

Torsional modulus of elasticity is about one-third to one-half the tension modulus.

of the cylinder, and then the thickness should just equal that of the walls.

If the head has to be made flat the formula for the thickness when well supported by bolts is

$$t=0.55d\sqrt{\frac{p}{S}},$$

for a round plate (d=diameter), while

$$t=0.35a\sqrt{\frac{p}{S}}$$

holds for a square plate (a=side of square), and

$$t=0.5\frac{ab}{(a^2+b^2)^{\frac{1}{2}}}\sqrt{\frac{p}{S}}$$

for a rectangular plate of length a and width b.

For manhole or handhole covers which are of elliptical form, the thickness is fixed by the formula

$$t=0.75\frac{ab}{(a^2+b^2)^{\frac{1}{2}}}\sqrt{\frac{p}{S}}.$$

In all cases the endeavor should be made to eliminate flat places, and when such must exist they must be designed for thickness as outlined above. At times ribs are used to stiffen flat places, and at times stay bolts are used crossing from side to side.

The thickness of heavy-pressure pump cylinders is determined by either one of two formulæ, Barlow's:

$$t=\frac{pr_2}{S-p},$$

or Lamé's:

$$t=\frac{r_1^2+r_2^2}{r_1+r_2}\frac{p}{S};$$

t=thickness in inches;
p=hydrostatic pressure in lbs. per sq.in.;
r_1=the outside radius of cylinder, in inches;
r_2=the inside radius of cylinder in inches;
S=maximum stress in lbs. per sq.in.

In designing the cylinders for hydraulic presses or pressure pumps the metal thickness should be uniform, as thicker parts of castings cool after the other parts have solidified. There is danger, therefore, that cooling strains will be set up and cracks may develop.

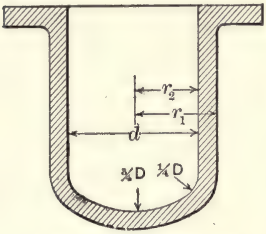


FIG. 273.—Pressure Cylinder.

There should be no sudden changes in the thickness of the section and all corners must be well filleted. These points should be observed in all cylinder design or in any other castings.

When the core of the heavy cylinder castings has to be supported at each end a hole must be left at the bottom of the cylinder, and when such is the case it is closed by driving in a plug or by using a packed joint (Fig. 274) in which a hydraulic packing of some form is used.

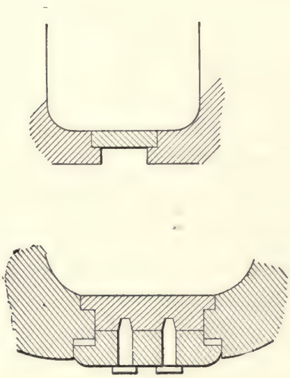


FIG. 274.—Core Hole Cap.

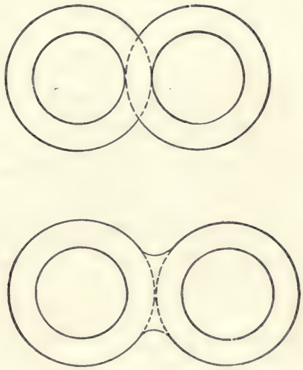


FIG. 275.—Twin-Cylinder Casting.

When two hydraulic cylinders are to be used in one casting, there must be sufficient metal where the two cylinders come together to support the total load in each cylinder. The upper part of Fig. 275 shows the incorrect method, while in the case below there is sufficient metal to carry the load.

When there are to be openings made into cylinder walls

for pipes, manholes, or other purposes, the edge of the hole should be reinforced by a ring or boss, and the amount of metal added should equal or should exceed the amount of metal removed in the section, as shown in Fig. 276. In this figure the amount of metal cut away is dt square inches, and this

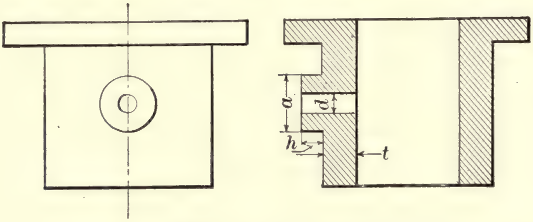


FIG. 276.—Branch Connection Reinforcement.

should be equal to or less than $(a-d)h$. These openings should be avoided, but when necessary in every case the reinforcement should be made.

FLANGES

The flanges to be used on cylinders, heads, manhole covers, and covers for other parts of the pumps are in general made equal to 3 diameters of bolt in width; this gives $1\frac{1}{2}d$ from the center of bolt to edge, a proportion which is common in all design work. Until the size of the bolts is determined this cannot be fixed. It is best, therefore, to assume the diameter of bolt, and design the number to be used. The bolts should be taken as large as possible. Small bolts are apt to be twisted off. It is well to use nothing less than $\frac{5}{8}$-inch bolts except when very small parts are used. On cylinders from 8 to 15 inches, $\frac{3}{4}$-inch bolts should be assumed; from 15 to 24 inches, 1 inch; and for larger sizes, the diameter should be more than 1 inch.

The flange width is now made equal to $3d$.

The thickness of the flange, t'', is made 1.2 to $1.4t$; where $t=$ thickness of cylinder walls.

The bolts support the pressure on the area A of the

cover exposed to the water pressure. The number n is given by

$$n=\frac{pA}{S\frac{\pi d^2}{4}}=\frac{pA}{Sa};$$

a = area of bolts at root of thread found in table of bolts in sq.in.;
S = allowable stress in lbs. per sq.in.;
A = area supported in sq.in. (usually to the inner edge of cover, although if packing is not tight this area may be increased);
p = water pressure in lbs. per sq.in.

After finding the number of bolts, the pitch of the bolts should be found, and if this is greater than

$$40\sqrt{\frac{t''}{p}},$$

where t'' is the thickness of the flange, more bolts must be used. The quantity above is the maximum allowable pitch for a tight joint.

Water Pistons

The water piston is made up of various parts, but it may be considered as a flat plate supported at the center and loaded uniformly. In this case the thickness would be about two-thirds the thickness of the cylinder head. If the piston is cored, as in Fig. 297, Chapter VIII, this thickness is made less, as the casting acts like a deep beam. The thickness of each part could be made less than one-third the thickness of the head but this amount may be used. If the two parts are separate, as in Fig. 199, then each should be of the full thickness.

Piston design is in most cases empirical and the proportional number is $\frac{D\sqrt{p}}{100}$ for cast iron or bronze. This will be considered in Chapter VIII, and until that place the proportions will not be given.

The packings used on water cylinders is often made of square-plaited flax, hemp or cotton, with a rubber core, indurated with rubber or alternate layers of rubber and cotton. The sizes are $\frac{1}{2}\times\frac{1}{2}$ inch, $\frac{5}{8}\times\frac{5}{8}$ inch, $\frac{3}{4}\times\frac{3}{4}$ inch, etc. In putting this in, the packing should be cut so that the ends do not touch by about the thickness of the packing. This packing swells on being wet. The depth of the packing is 0.4d on Fig. 195.

Piston Rods

The piston rod is designed to take compression only in a single-acting pump, and to take compression and tension in a double-acting pump. In many cases these rods are of steel, but in some small pumps where corrosion is to be avoided, phospor bronze, Muntz metal, or some other form of non-rusting metal is used.

Tobin bronze is an alloy of copper (59%), zinc (38.4%), tin (2.16%). Iron (0.11%) and lead (0.31%) are sometimes found in the mixtures. It has a tensile strength as high as 75,000 when rolled, and at times 100,000 pounds per square inch, has been obtained after cold rolling. At a cherry red this bronze can be stamped and forged.

Muntz metal is an alloy of 3 parts of copper with two of zinc. It has a strength of about 50,000 pounds per square inch.

The piston rod is treated as a column in all cases, as it is the compressive load which fixes the size.

The figures of this chapter illustrate the method of attachment of the piston rod. It will be seen that the end of the rod has a taper extending from a shoulder or collar to a cylindrical-threaded portion. The threaded portion takes plain tension; the shoulder and the tapered portion take the compression. It is well in forming the shoulder or collar to fillet the corner instead of leaving it absolutely square, as this fillet increases the strength, although it is not so easy to get a fit of the rod to a rounded corner.

The load on the rod is

$$P=\frac{\pi d^2}{4}p.$$

The area at the root of the thread is

$$a = \frac{P}{S_t}.$$

The diameter of this thread is found from a table of standard threads for the area a.

The rod is designed by a column formula:

$$\frac{P}{\frac{\pi d^2}{4}} = \frac{S_c}{\left[1 + K\frac{l^2}{r^2}\right]}, \text{ Rankin's Formula;}$$

$$\frac{P}{\frac{\pi d^2}{4}} = S_c - K'\frac{l}{r}, \text{ Straight-Line Formula;}$$

P = total load on one rod in lbs;
d = diameter of rod in inches;
r = radius of gyration $= \frac{d}{4}$;
S_c = allowable working stress in lbs. per sq.in.;
l = greatest length of rod between supports in inches;
$K = \frac{1}{6250}$ for steel;
$K' = 225$.

These equations must be solved by trial by first getting the value of d from $\frac{P}{\frac{\pi d^2}{4}} = S_c$, and then using this to find r; solve for d after the approximate value of r has been inserted.

The shoulder used on the rod should be $\frac{1}{8}$ to $\frac{1}{4}$ inch on small rod, while $\frac{1}{2}$ inch may be used if a collar is formed. The taper part of the rod should reduce the section to the diameter at the thread. A taper such that the diameter is reduced 3 inches in a foot is a good one to use.

Stuffing Boxes

The stuffing-box design has been given earlier in this chapter.

Valves and Springs

The proportion of valves will be seen from the various figures. The diameter of the valves has been discussed in Chapter V. The spring design will now be considered. The formulæ for springs are:

$$P=\frac{\pi d^3S}{16r} \qquad (a);$$

$$\lambda=\frac{64nPr^3}{E_sd^4} \qquad (b);$$

where S = safe shearing strength in lbs. per sq.in.;
P = load in lbs.;
n = total number of coils;
d = diameter of wire in inches;
E = torsional modulus of elasticity;
λ = compression in inches.

From formula (a) the radius of coil may be found to carry the load for an assumed diameter and stress for the spring wire. With this radius the number n may be found to give the necessary compression with a given change of load.

$$\lambda-\lambda'=\frac{64nr^3}{E_sd^4}(P-P').$$

The pitch of the helix is found by assuming the height allowable, and putting n turns in this height. If this result is not thought proper a different diameter of wire may be taken; a new r and n, found. This should be tried until one of the proper form is found. Although r and n are definite and would give the desired result, they do not always fit the other parts of the design.

The following proportions for valves are used in practice, according to Barr:

VALVE DISCS

Diameter.	Thickness.	Hole.
2″	$\frac{3}{8}$″	$\frac{1}{2}$″
$2\frac{1}{2}$″	$\frac{7}{16}$″	$\frac{1}{2}$″
3″	$\frac{1}{2}$″	$\frac{9}{16}$″
$3\frac{1}{2}$″	$\frac{5}{8}$″	$\frac{5}{8}$″
4″	$\frac{5}{8}$″	$\frac{11}{16}$″
$4\frac{1}{2}$″	$\frac{3}{4}$″	$\frac{13}{16}$″
5″	$\frac{3}{4}$″	$\frac{13}{16}$″

Taper of thread on brass valve seat to fit valve deck 1 inch per foot. Valve stem $\frac{1}{16}$ inch less than diameter of hole. Plate on top of valve $\frac{1}{32}$ inch thick, and of diameter equal to three-fourths diameter of disc. Plate $\frac{1}{16}$ inch thick for valves $4\frac{1}{2}$ inches and over. Springs are made of the following sizes:

Diameter of Valve.	Size of Wire.
2″	No. 12
$2\frac{1}{2}$″	No. 12
3″	No. 10
$3\frac{1}{2}$″	No. 10
4″	No. 8
$4\frac{1}{2}$″	No. 8

Springs are usually coiled with a diameter of one-half the valve diameter, and with five turns they have sufficient elasticity.

Fig. 277 illustrates the method of installing a small modern pump and applies the general principles outlined in these two chapters. The foot valve *A* is applied at the lower end of the suction pipe. The suction air chamber *B* is applied so as to receive directly the impulse from the water. The strainer at *C* is conveniently placed for cleaning. A check valve *D* permits one to relieve the valve chamber of pressure when necessary to examine the valves without draining the discharge main. The priming pipe *E* is used to prime the pumps. By opening the waste-drain pipe *F* and the primer *E* the pump is filled with water, the air being driven out. The waste pipe may be opened in starting the pump so that air may be removed before the full pressure is exerted. Such an arrangement may be

necessary in starting a compound pump where pressure is needed on both pistons to start the pump against full head. In such a case the pump could not start unless live steam were by-passed into the low pressure cylinder. Another method sometimes used, which is the equivalent of this, is to connect the two ends of the water cylinder by a cross connection to eliminate water pressure. After both cylinders have received the proper

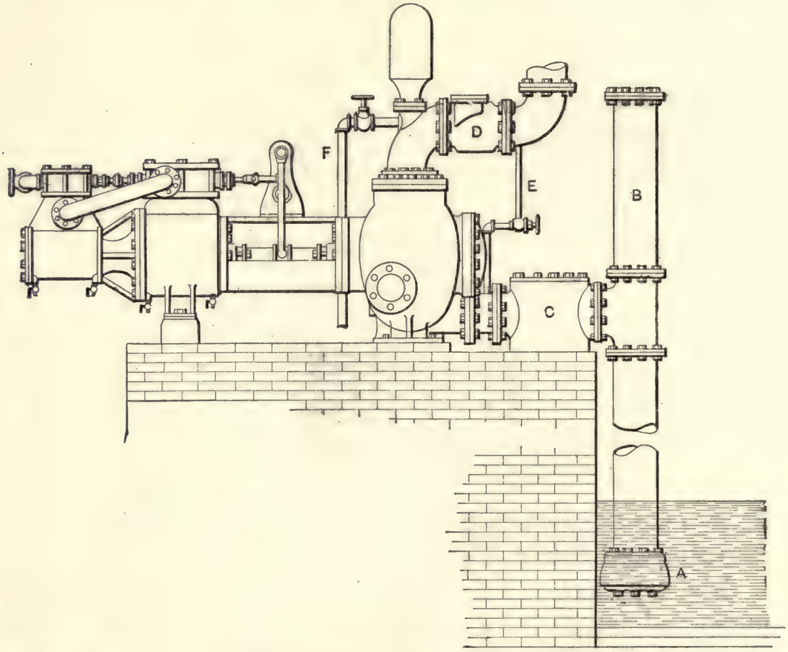


FIG. 277.—Arrangement of Pump.

steam on finishing a few strokes the waste or the connection between the ends is closed, and the pump discharges against the working head.

The figure illustrates large-size suction pipes and short, direct connections, so necessary for proper action. Care must be taken in making up these joints to have them air tight.

The following quotations from the catalogue of the Snow Pump Works is valuable, although many of them have been given previously in this chapter:

"INFORMATION CONCERNING PIPE CONNECTIONS

"Fig. 277 shows in a general way the proper method of piping a pump. Faulty connections are generally the cause of the improper action of a pump, and great care should, therefore, be taken to have everything right before starting. To accomplish this, note carefully and understand thoroughly the following:

"Be sure that the quantity of water you desire to pump is available and that your pump is within easy reach of it when it is at its lowest level.

"Locate your pump as near the source of suction supply, both vertically and horizontally, as is possible or convenient; but never place it in such a location that the sum of the following three items would exceed a total of 26 feet:

"1. Height in feet from the discharge valves of the pump to the lowest level of the surface of the suction water.

"2. Total frictional loss in suction pipe in feet head.

"3. Total frictional loss in feet head, due to elbows and tees (assumed as being the equivalent of the frictional loss due to 100 feet of same size of pipe, for each elbow or tee).

"EXAMPLE.—Would a pump having an easy capacity of 750 U. S. gallons per minute operate satisfactorily at this capacity if the height from the delivery valves to the surface of the suction supply was 20 feet, the suction pipe 8 inches diameter, and 800 feet long, and having two 8-inch standard elbows?

"ANSWER.—No. (Ascertained as follows):

"Height from delivery valves to surface of water . . . =	20. ft.
"Total friction in 800 ft. of 8-in. pipe ($=.53\times8\times2.31$) =	9.79 ft.
"Total friction in two 8-in. elbows ($=.53\times2.31\times2$) =	2.45 ft.
Total	32.24 ft.

"The sum of these three items is in this case about 32 feet, or 6 feet greater than the 26 feet above mentioned. Therefore, the pump should be lowered 6 feet, or the frictional resistance

in the suction pipe reduced by about 6 feet, by increasing the size of the suction pipe.

"Lay your suction pipe so that it slopes away from the pump gradually. A suction pipe should have no air pockets in its entire length, but should be so laid that if air be admitted to it, near the intake end, with the pump standing still, the air would rise to the pump or suction air chamber, and not be pocketed in some high part of the suction pipe. A slope of 1 per cent (1 foot drop in 100 feet of length) will be found very satisfactory.

"Be sure that your suction piping is absolutely tight, for a very small air leak will cause a pump to work improperly. The suction pipe should be tested with about 20 pounds water pressure after it has been laid and before it is covered. If the test shows up a leak, fix it; it is not 'good enough.'

"Keep the end of your suction pipe well under water. It should never have less than 3 feet above it and 6 or 8 feet would be much better.

"When you take suction from a tank into which the returns from hydraulic elevators or presses empty, take care that the returns enter the tank as far away from the suction-pipe opening as possible, for the pump is liable to get air if the returns empty near the suction opening.

"If two or more pumps draw from the same suction pipe, or if water comes to the pump under a head, a gate valve should be placed on the suction pipe of each pump, to enable you to open up any one pump cylinder for repairs or examination without interfering with the operation of the other pumps. We recommend on larger sizes, when practicable and when not too costly, that each pump have a separate individual suction line entirely independent of the suction line of any other pump.

"A suction air chamber will be found desirable in all cases, and indispensable in cases where the sum of the three items referred to in a previous paragraph exceeds 10 feet, or when the suction pipe is long.

"A foot valve is desirable in all cases (except when suction

water comes to the pump under a head) and indispensable when the suction lift exceeds 10 feet. By its use the pump and suction pipe are kept primed when the pump is shut down, and permits of easily priming the pump and suction pipe if purposely emptied, thus enabling the pump to be easily started at any time.

"In all cases where the water contains sticks, weeds, rags, or other rubbish a strainer should be used on the suction pipe, to prevent them from getting into the pump and clogging valves and passages. If a foot valve is used, a strainer placed outside the foot valve is the best; but if no foot valve is used, a box strainer, placed near the pump, and so designed that by removing the strainer cover all accumulations can be removed, will be found the most desirable. Keep the strainer clear from accumulation of rubbish.

"When a foot valve is used, a drain valve should be placed near the surface of the water, to enable the suction pipe to be drained when desired.

"A relief valve, set to blow at about 20 pounds pressure, should also be placed on the suction pipe near the pump, to prevent delivery pressure, if over 50 pounds, from accumulating in the suction chamber of the pump or the suction pipe. This does not cost much and may sometime save you the cost of replacing a broken pump cylinder or foot valve, due to carelessness.

"In cases where the pump gets its supply directly from driven wells, it will be found most satisfactory to place a large tank on end in some part of the pump house, and to run the pipes from the wells into the side of the tank not far from the bottom. Take the suction for your pump from the bottom of the tank, and lead a small pipe from the top of the tank to an air pump provided for the purpose, or to the condenser of the main pump, if its air pump is large enough. A gauge glass on the side of the tank will show the level of the water in same, and by opening the valve on the small air pipe a certain amount of the air may be removed from the tank and the pump will then get no air, but pump its full displacement of

water. The air may be extracted automatically from the tank by means of a float arrangement.

"A check valve D on the discharge pipe will be found very convenient.

"A gate valve should always be placed on the discharge pipe outside of the check valve.

"A priming pipe should always be connected from the discharge pipe, outside of the gate, to the suction pipe, if a foot valve is used. This will enable the pump cylinders and suction piping to be primed, if empty, before starting. If you have no suitable relief valve on the suction pipe, be very careful, in priming with this pipe, that you do not let delivery pressure accumulate in the suction pipe. This will be prevented by having the starting waste valve F open before you open the priming pipe valve E. This should always be open before starting your pump (whether you have a foot valve or not), as by this means the pump is enabled to discharge the air from the pump cylinders and suction pipe through this starting valve against a light pressure. As soon as water is discharged through the starting valve, shut it and open your steam throttle valve, and the pump will then discharge through the discharge main, opening the check valve automatically. If you have a foot valve or a gate on your suction pipe, and no relief valve, be careful to open the starting valve at the instant you shut the pump down and leave it open until after you have started again, as by so doing you prevent the possibility of pressure accumulating in the suction. The pet cock in the force chamber of small-size pumps is intended to be used in the same manner as the starting valve above referred to.

"When shutting the pump down in late fall, winter, or early spring, be sure and open all steam and pump-end drains, and leave them open, otherwise your steam or pump cylinders or other parts may be cracked, due to water freezing inside of them, and you may be forced to purchase practically a new pump. A little care will prevent this occurrence.

"Do not try to raise hot water, crude petroleum, or any

thick liquids by suction, but wherever possible have the liquid flow to the pump under a suction head.

"Keep the steam cylinders and valve motion of your pump well lubricated. Oil is cheaper than repair parts.

"Do not pack the stuffing boxes too tightly, and do not let the packing stay in until it gets hard and cuts the piston rods or plungers. Renew it sufficiently often to keep it soft and pliable. If the pump runs badly, make sure that the pump valves, packed pistons, or plungers, and suction and discharge connections are all right before examining the steam end."

CHAPTER VII

DYNAMICS OF STEAM END

In the design of a pump the variation of pressure has been found together with the size of the pump cylinders, and it is now necessary to find the size of the steam cylinders to operate the water end, the size of the electric motor to drive the pump, or the size of the water wheel which is required.

The curves constructed in Figs. 197 and 198 can be combined to give the indicator card of the head end of the pump. The crank-end card of the water end, if the pump were double acting, would be found in a similar manner. From this combined card (Fig. 278), which is the indicator card, the power to be given to the water may be computed.

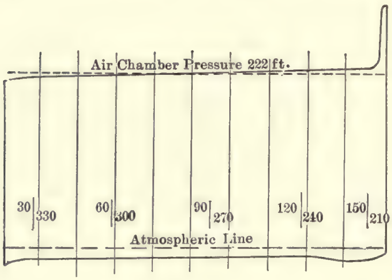


Fig. 278.—Pump Card.

Assume that there are three single-acting pumps giving cards similar to Fig. 278 connected together, the discharge pipes of each cylinder being 18 inches in diameter, and continued separately to the height considered in Chapter V. The area of the card then would give the work done on the water. The mean height of the card is 235.2 feet or 102.1 pounds per square inch. This includes all friction of the water, and would give the power required if an allowance is made for the mechanical efficiency of the pump.

The water indicated horse power is

$$\text{I.H.P.} = \frac{3PLAN}{33{,}000};$$

where P = mean effective pressure in lbs. per sq.in.;
L = stroke in feet;
A = area of piston in sq.in.;
N = number of revolutions per minute;

$$\therefore \text{ I.H.P.} = \frac{3 \times 102.1 \times \frac{30}{12} \times 314 \times 60}{33,000} = 437.2.$$

The mechanical efficiency of the water ends of pumps will vary with the construction and condition of packing. With a plunger pump an efficiency of 97 per cent is not uncommon; and this will be taken as the mechanical efficiency in the problem studied. With inside-packed pumps the friction may increase so that for such pumps 92 per cent will be used. These values will vary, and if the glands or packing rings on pistons are tightened too much these values will be smaller.

The mechanical efficiency of the complete steam-driven unit, including the power for the air pump of the condenser on large pumps is about 95 per cent, while 90 per cent or even 80 per cent should be used for small pumps. With a gear-driven pump the gears will absorb about 5 per cent of the power per pair, so that with a back gear the loss of the gearing amounts to 10 per cent. The efficiency of engines and electric motors may be taken as 90 per cent to 95 per cent, and the efficiency of water turbines as 80 per cent. Using the figures above, the indicated horse power of the engine for driving, the electrical power and the water horse power applied to the wheel will be respectively:

$$\text{I.H.P.} = \frac{437.2}{.95} = 460 \text{ I.H.P.};$$

$$\text{K.W.} = \frac{437.2 \times .746}{.97 \times .90 \times .95} = 527 \times .746 = 394 \text{ K.W.};$$

$$\text{W.H.P.} = \frac{437.2}{.97 \times .90 \times .80} = 626 \text{ W.H.P.}$$

To carry further the design of the steam end for the purpose of finding the size of fly wheels or other parts it will be assumed that this pump of three cylinders is to be driven by a triple-

expansion engine, with three cylinders, one of them in tandem with each of the water cylinders. The plungers are single acting, and it will be assumed that they are built solid to give a greater pressure on the downward stroke.

To find the size of the engine, assume that the steam expands in one cylinder, and that the theoretical card is of the form shown in Fig. 279. This card assumes no clearance and compression, the effect of these together being equal to a decrease in area of about 3 per cent. If the initial absolute pressure

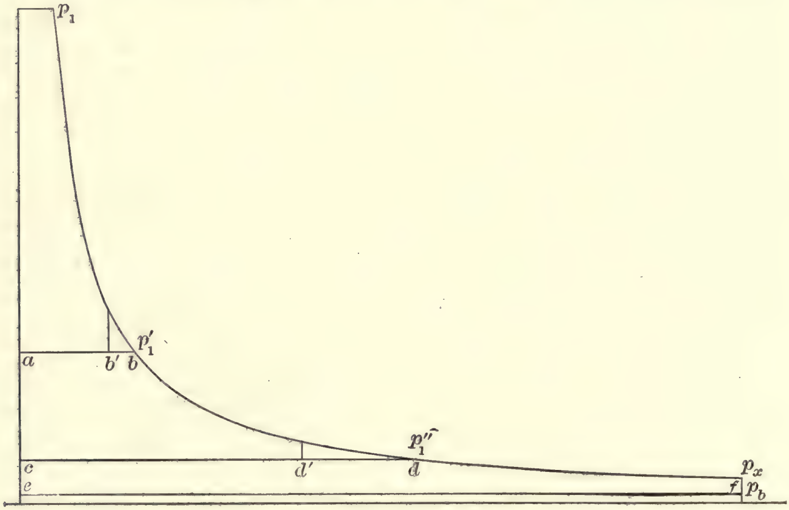


FIG. 279.—Indicator Cards with no Clearance.

be called p_1, the pressure at the end of expansion p_x, the back pressure p_b, and if the curve of expansion be taken as a rectangular hyperbola, the mean height of the figure becomes

$$\text{Mean height} = \frac{p_1}{\frac{p_1}{p_x}}\left[1 + \log_e \frac{p_1}{p_x}\right] - p_b$$

$$= p_x\left[1 + \log_e \frac{p_1}{p_x}\right] - p_b.$$

This quantity is multiplied by 0.97 to allow for the effect of clearance and compression. There is another effect for

which allowance must be made. This is the effect of the valve gearing on the indicator card. The corners of the card are rounded, due to slow valve action, and there is some change of pressure between the various cards as shown in Fig. 280. This means that a diagram factor must be applied to give the proportion of the theoretical card which may be actually obtained. The value of this will be taken 95 per cent for D-slide valve engines, and 97 per cent for Corliss engines.

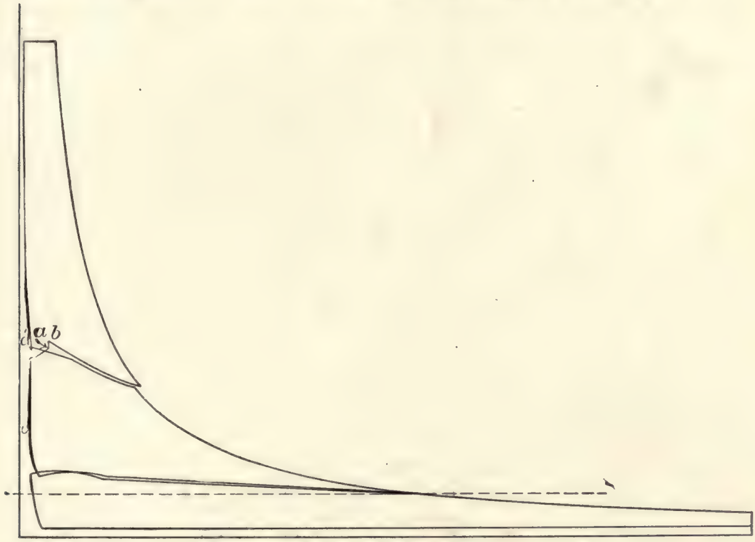


FIG. 280.—Combined Cards.

The probable mean effective pressure will then be given by

$$\text{M.E.P.} = \text{diagram factor} \times .97 \times \left[p_x\left(1 + \log_e \frac{p_1}{p_x}\right) - p_b\right].$$

The size of the low-pressure cylinder will be found by the formula

$$\text{I.H.P.} = \frac{2PLAN}{33{,}000}.$$

In this formula the values of P, L and N for the tandem-steam cylinder are known at this point, and A may be found. If the engine is to drive by gears $2LN$ may be assumed, next

N, after which L and A are computed. In some cases a ratio of L to D is assumed. Having the size of the low-pressure cylinder the size of the other cylinders may be assumed by taking the ratio of cylinder volumes from practice. A better way is to divide the area of Fig. 279 into three equal parts, giving equal work to each cylinder. After this the sizes of the cylinders are found from the figure.

If the receiver is assumed to be very large the pressure between stages will be constant.

If p'_1 is the initial pressure in the intermediate cylinder, and p''_1 that in the low-pressure cylinder, the following formula may be used to find these, assuming equal work:

$$\tfrac{1}{3}\left\{p_x\left[1+\log_e\frac{p_1}{p_x}\right]-p_b\right\}V_x=\left\{p_x\left[1+\log_e\frac{p''_1}{p_x}\right]-p_b\right\}V_x,$$

for low pressure;

$$=\tfrac{1}{2}\left\{p_x\left[1+\log_e\frac{p'_1}{p_x}\right]-p_b\right\}V_x,$$

for low and intermediate together.

From these:

$$\log_e p''_1=\tfrac{1}{3}\left[\log_e p_1 p_x^2+2\left(\frac{p_b}{p_x}-1\right)\right];$$

$$\log p'_1=\tfrac{1}{3}\left[\log_e p_1^2 p_x+\left(\frac{p_b}{p_x}-1\right)\right].$$

For

$$p_1=150.3 \text{ lb. gauge},$$
$$p_x=-6.7 \text{ lb. gauge},$$
$$p_b=-11.7 \text{ lb. gauge},$$

the following pressures result:

$$p'_1=50.4 \text{ absolute};$$
$$p''_1=14.47 \text{ absolute}.$$

These values have been marked on Fig. 279. From these, the relative dimension of the cylinders may be found, since the lines ab, cd, and ef represent respectively the volumes of the

high-pressure cylinder, of the intermediate-pressure cylinder, and of the low-pressure cylinder, hence

$$\frac{V_{\text{h.p.}}}{V_{\text{l.p.}}}=\frac{ab}{ef};$$

$$\frac{V_{\text{i.p.}}}{V_{\text{l.p.}}}=\frac{cd}{ef}.$$

These ratios are the theoretical ratios of the cylinders if the expansion is complete in each cylinder. If there is free expansion in the different cylinders, the proportionate drops in each are assumed, and then the points b and d are moved to b' and d', and the ratios above are changed to $\frac{ab'}{ef}$ and $\frac{cd'}{ef}$. With such ratios the volume of the receiver and piping is expressed in terms of the cylinder volumes from practice as are the clearance volumes. So soon as the low-pressure cylinder volume is computed from the horse-power equation, the volumes of the other parts are all known and the combined individual cards may be drawn as in Fig. 280. The receiver and piping between each cylinder is taken as about 250 per cent of the cylinder from which the steam is discharging, and the clearance on the three cylinders, to be 5 per cent for the high-pressure cylinder, 2.8 per cent for the intermediate-pressure cylinder, and 1.5 per cent for the low-pressure cylinder. These values are given as guide so that these quantities may be computed, but a designer after making a number of designs would use the tables of proportions which he had computed from previous experience.

The value of a reheater is questioned, a matter which will be taken up later.

With the values above, the data for the problem considered give the following results:

$$\text{M.E.P.}=.96\times .90[8(1+\log_e \tfrac{165}{8})-3];$$
$$=25.2 \text{ per sq. in.};$$

$$\text{I.H.P.}=460=\frac{2\times 25.2\times A\times \frac{30}{12}\times 60}{33{,}000}.$$

$$A = 2020 \text{ sq.in.};$$
$$D = 51 \text{ in.};$$
$$2LN = 2 \times \tfrac{30}{12} \times 60 = 300 \text{ ft. per min., which is not too high};$$

$$\frac{V_{\text{h.p.}}}{V_{\text{l.p.}}} = \frac{8}{50.4} = \frac{1}{6.3} \text{ (since } pv = k.);$$

$$\frac{V_{\text{i.p.}}}{V_{\text{l.p.}}} = \frac{8}{14.47} = \frac{1}{1.81};$$

$$\frac{V_{\text{h.p.}}}{V_{\text{i.p.}}} = \frac{1.81}{6.3} = \frac{1}{3.48};$$

$$\therefore\ V_{\text{h.p.}} : V_{\text{i.p.}} : V_{\text{l.p.}} = 1 : 3.48 : 6.3.$$

The strokes are all the same, hence

$$D_h^2 : D_i^2 : D_l^2 = 1 : 3.48 : 6.3,$$

Since

$$D_L = 51 \text{ in.};$$
$$D_h = 20.33 \text{ in. and}$$
$$D_i = 37.8 \text{ in.}$$

The first approximation of cylinder sizes will therefore be

20×30
37×30
51×30

The clearance volumes are

470 cu.in.
880 "
890 "

The receiver volumes are

23,530 cu.in.
81,884 "

From these volumes and pressures the theoretical indicator cards can be drawn as in Fig. 280. The complete expansion of steam to the receiver pressure is found in most pumping engines, and for this reason these cards have been constructed in this manner. The cranks are assumed at 120°, and the pressures and volumes of admission to the various cylinders have been

constructed graphically. The receiver volumes and clearance volumes have been used in constructing curves where possible; where not possible, the pressures have been computed. As an example, the point to which the letter *a* (Fig. 280) refers, has been computed by assuming the product of a final p and v as equal to the sum of the original products, pv, before mixing,

$$p_a(v_{ah}+v_{aI}+v_R)=p_b(v_{bh}+v_R)+p_c v_{cI}.$$

The pressures in the receivers have been made the terminal pressure of expansion, and the exhaust has been arranged to give this result.

After the cards have been drawn they have been used to construct cards 4 inches long, and to various scales. The cards drawn are seen to be of different size from the original card with no clearance; the H.P. card has been reduced, the L.P. card increased, and there is little change in the I.P. card. To make certain that the steam cylinders are of sufficient size the I.H.P. of the total engine should be found in terms of the L.P. cylinder displacement, assuming that the volumes of the cylinders are of the ratios given originally. Allowance is also made for the piston rod by considering it to be $2\frac{1}{2}$ per cent of the piston area. After finding the M.E.P. from each theoretic card in Fig. 280 or Fig. 281, the following formula may be used:

$$\text{I.H.P.}=\frac{1.975}{33,000\times 12}\left(P_h\frac{1}{6.3}+P_i\frac{3.48}{6.3}+P_L\right)(\text{Disp})_{\text{low}}N.$$

From this a new size of L.P. may be computed.

The M.E.P. from the indicator cards give 47 pounds for the high-pressure cylinder, 15.9 pounds for the intermediate, and 8.2 pounds for low-pressure cylinder. Substituting these, the equation becomes

$$460=\frac{1.975\times 60}{33,000\times 12}\left(\frac{47}{6.3}+\frac{3.48}{6.3}15.9+8.2\right)D;$$

$$D=56,220 \text{ cu.in.};$$

$$\text{Area}=\frac{56,220}{30}=1874 \text{ sq.in.};$$

$$\text{Diameter}=49 \text{ in.}$$

Hence

$$\text{Diameter intermediate} = 36.4 \text{ in.};$$
$$\text{Diameter high} = 19.5 \text{ in.}$$

Use 50, 37 and 20. It would be well at this time to redesign the cylinder and cards of Fig. 280 so as to make the work on each cylinder the same.

The areas of these cards together will be greater than the area of the water cards by the amount of friction. The friction of the steam and water pistons and the friction of the stuffing box will be considered to be three-fourths of the total friction, and the remaining part will be taken uniformly by the bearings of the shaft.

This gives 22.8 H.P. (460 − 437.2) as the total to be absorbed, of which 17.1 H.P. is used in the packing and 5.7 in the journals.

17.1 H.P. means $\frac{17.1 \times 33,000}{60} = 9405$ ft.lbs. per revolution. The mean pressure from this over $2\frac{1}{2}$-foot stroke would be $\frac{9405}{5 \times 3} = 627$ pounds for each unit above and the same amount below the zero of pressure, as this force operates to require work and always to hinder motion. This quantity is to be divided by the areas of the pistons in order to reduce this to the same basis as that used in the diagrams.

The weights of the various parts will be assumed to be those given in the table below. The weights would ordinarily be determined from the design of the piston, piston rod, cross-head, and connecting rod, which would be made before proceeding with this part of the work. For their design, see Chapter VIII.

Weight of H.P. piston	225	lbs.
Weight of I.P. piston	900	"
Weight of L.P. piston	1400	"
Weight of each piston rod	125	"
Weight of each cross-head	800	"
Weight of each connecting rod	350	"
Weight of 4 rods to plunger from cross-head	450	"
Weight of plunger	5000	"

This gives as the sum of the weights of the reciprocating parts of each cylinder the following:

High-pressure unit.	6900 lbs.
Intermediate-pressure unit.	7575 "
Low-pressure unit.	8075 "

Reducing these to pressure per square inch of specific steam cylinder these values become:

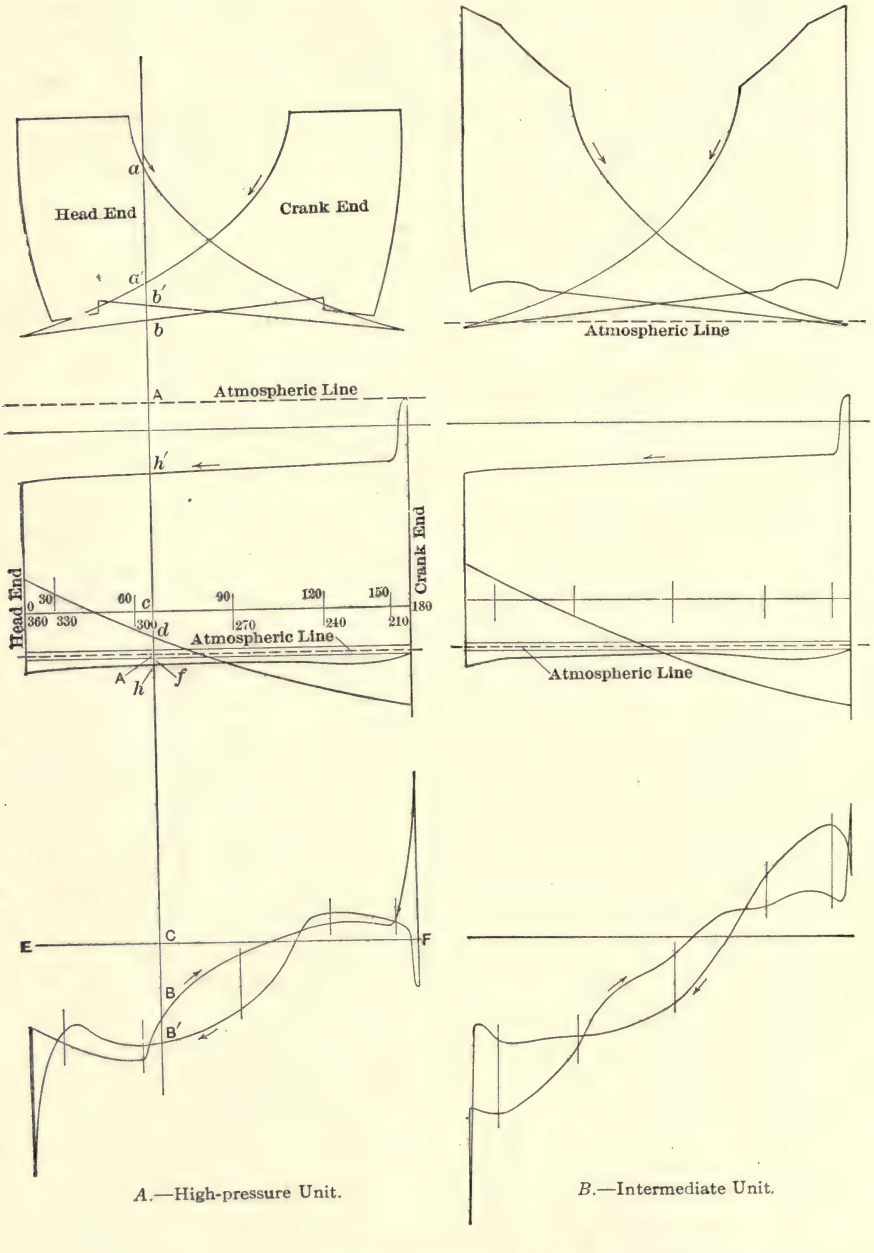
High-pressure piston 22 lbs. per sq.in.
Intermediate-pressure piston, 7 lbs. per sq.in.
Low-pressure piston, 4.1 lbs. per sq.in.
Friction H.P. piston = 2.0 lbs. per sq.in.
Friction I.P. piston = 0.575 lb. per sq.in.
Friction L.P. piston = 0.318 lb. per sq.in.

Taking the indicator cards from the separate steam and water cylinders, as shown in Fig. 281, it is necessary to add 22 pounds per square inch for the high-pressure unit on the down stroke, to get the effective pressure, and subtract the same on the up stroke. Moreover, the back steam pressure on the crank end must be subtracted from the head end on the down stroke in order to get the net pressure from the steam. As these piston areas are not the same on each end of a piston, to reduce them to the same scale the crank-end pressures should be multiplied by $\frac{A_c}{A_h}$, where area A_c is the area of the crank end of the cylinder and the area A_h is the area of the head end. In the case of a 20-inch cylinder with a 3-inch rod, this ratio is 0.98, and hence it may be considered as unity. On the other pistons it is still closer to unity.

The reciprocating parts require accelerating and therefore a force equal to

$$\frac{w}{g}a = \frac{w}{g}\omega^2 r\left(\cos\theta + \frac{1}{n}\cos 2\theta\right)$$

is required on each square inch of piston area, if w represents the weight of the parts per square inch of piston area. The values of a for the different positions may be taken from Chapter

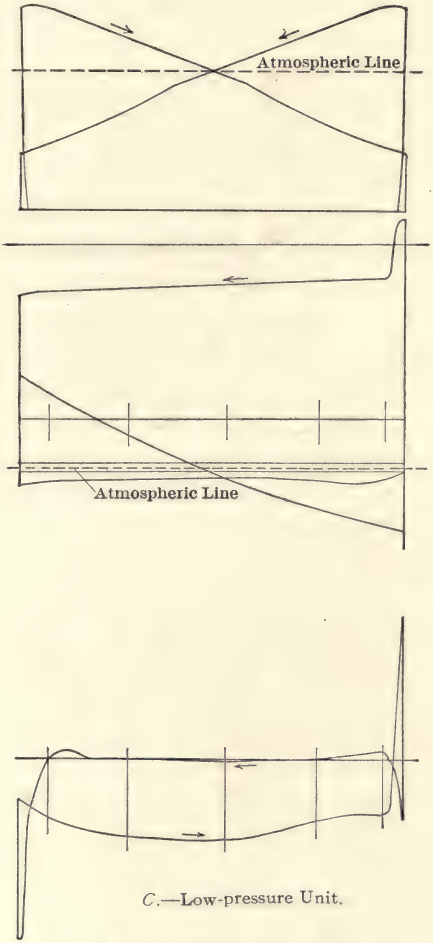


A.—High-pressure Unit.

B.—Intermediate Unit.

FIG. 281.—Combined Cards.

V, page 192, and plotted on stroke position after multiplying by $\frac{w}{g}\omega^2 r$ for each cylinder. These curves are plotted in Fig. 281. At the beginning of each stroke the inertia force must be



C.—Low-pressure Unit.

Fig. 281.—Combined Cards.

subtracted from the net steam force, while at the end of the stroke it is added. The curve repeats itself with the same numerical values, but of opposite signs for 210° as for 150°,

for 240° as for 120°, etc. Hence the curve need not be redrawn, but the same curve may represent each stroke, provided the sign of the quantities be changed for the two strokes.

Having now these various forces which act on the system, it is a simple matter to find out how much of it remains unbalanced and must therefore pass from this system into the others, or into the fly wheel by way of the connecting rod and shaft.

Take any point A on the high-pressure set of cards and consider this to be on the up stroke (head to crank), when the fly wheel and crank are assumed to be above the steam and water cylinders which are adjacent. The steam pressure on the upper side is aA, on the lower side bA. The weight is cA, the inertia is dA, the friction of packing is Af and the water pressure is hA below the atmosphere. The water pressure is measured from the atmosphere because the air pressure is pressing down on an area equal to the area of the plunger minus the area of the steam piston rod (a small quantity compared with the piston or plunger area) and this balances practically 14.7 pounds of the water pressure. The pressure on the water cylinder has been reduced to a figure which represents pressure per square inch of steam piston area, by redrawing the diagram of Fig. 278, using for pressures $P\frac{A_{\text{water end}}}{A_{\text{steam end}}}$, and using the same scale as used for the other diagrams. The net pressure remaining in the system per square inch of steam cylinder area is

$$P=aA-bA-cA-dA-Af-hA=CB.$$

On the down stroke at the point A the net pressure becomes

$$P_1=a'A-b'A+cA+dA-Af-h'A=CB'.$$

(Note that arrows showing directions of piston movement aid in this construction.) The figure is drawn by joining points for different positions. In the problem assumed, the work of each steam cylinder was taken about the same and that of each water cylinder was the same, hence the area of this resultant figure which represents the net work done by the

system outside of itself, must just equal its share of the work done in journal friction.

Since the probable cards are not arranged to give absolutely the same work on each cylinder, this is not quite true. The diagrams for the different cylinders have been made of the same length, but to make them of proper area, the high pressure was drawn with a scale of 50 pounds per square inch to the inch of height, the intermediate with $\frac{50}{3.48}$ or 14.38 pounds per square inch to the inch of height and the low pressure with $\frac{50}{6.3}$ or 7.95 pounds per square inch to the inch of height.

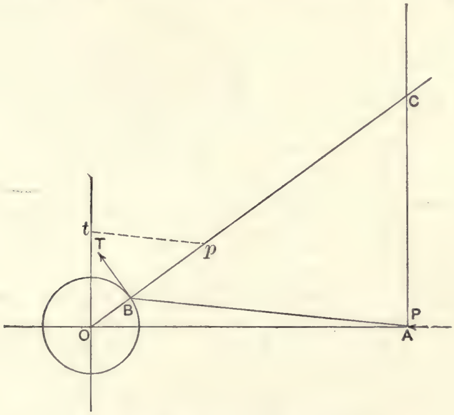


FIG. 282.—Tangential Effort Construction.

The cards, Fig. 281, have been constructed by laying off positive pressures above the base line *EF* on the crank head stroke, while for the head-crank stroke, positive pressures are laid off below. This gives a figure, the area of which represents the work or energy transmitted from this system to the others through the connecting rod and shaft.

Having now the force per square inch of piston which is transmitted to the crank by the connecting rod, the next operation is to find the value of the force produced by this in a direction tangent to the crank arm. In Fig. 282 the crank and connecting rod are shown by lines in a given position

of the crank. The end A of the rod is moving horizontally and therefore it is moving instantaneously about some point on a perpendicular to the path, as on AC for the position shown. The end B is moving in a tangent to the crank circle and therefore about some point in line with the crank radius OBC. Now

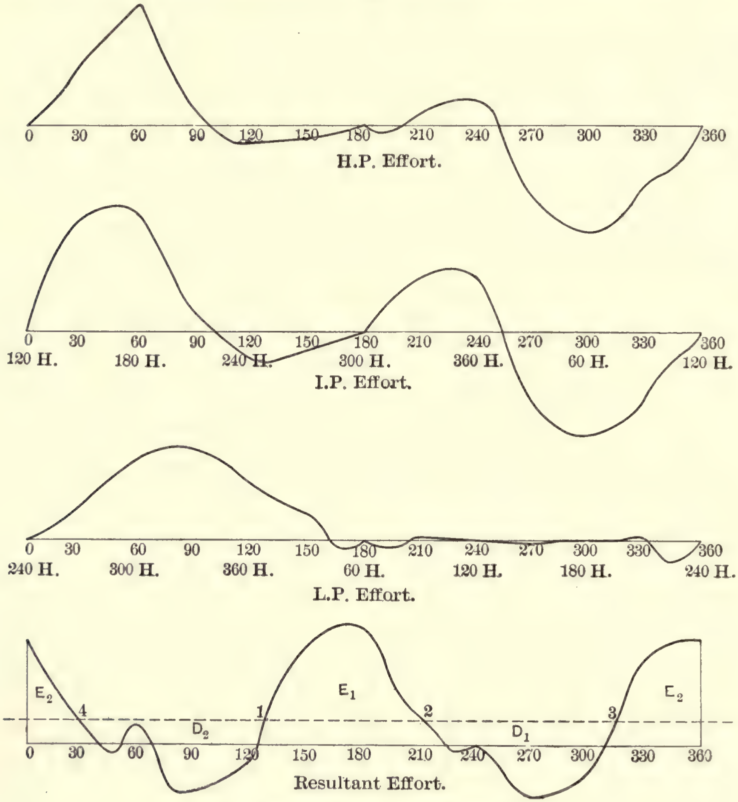


FIG. 283.—Tangential Efforts.

one end of the rod is moving about a point in one of the lines and the other end about a point on another line and the only point which satisfies both of these conditions is the point C. That is, for the instant considered the connecting rod is turning about the point C as much as if it were rigidly connected to a pivot at this point. If then it turns about this point, the force perpendicular to OB produced

by the piston force P, perpendicular to AC, is given by the equality

$$T\overline{CB} = P\overline{AC}.$$

The effect of inertia of the rod is such that this equation is not strictly true, but Jacobus has shown in a paper (A.S.M.E., XI, pp. 492 and 1116), that, for the design of the fly wheel, the approximate method of considering the connecting rod as having the motion of the piston and including it with the other reciprocating parts and neglecting the real inertia effect at this point, is sufficiently accurate.

To find the value of T graphically, lay off op equal to the pressure from the piston rod, and draw pt parallel to the connecting rod.

The triangles otp and CAB are similar, hence

$$\frac{CB}{AC} = \frac{Op}{Ot} = \frac{P}{Ot};$$

$$P\overline{AC} = \overline{Ot}\,\overline{AC};$$

$$P\overline{AC} = T\overline{AC}.$$

hence $$Ot = T.$$

For the piston pressure P at the cross-head the tangential effort T has been found. This is done for several points in Fig. 283 and plotted on a line representing the travel of the crank. The other two cards are drawn as shown.

The area scale of these figures, Figs. 281 and 283, is equal to the product of the two linear scales. The various scales are given below:

Card.	Length Scale.	Height Scale.	Area Scale.
H.P.	$\frac{5}{8}$ ft. per.in.	50 lbs. per sq.in. H.P. piston per in.	31.25 ft.-lbs. per sq.in. H.P. area per sq.in.
I.P.	$\frac{5}{8}$ ft. per in.	14.38 lbs. per sq.in.of I.P. piston per in.	8.99 ft.-lbs. per sq.in of I.P. area per sq.in.
L.P.	$\frac{5}{8}$ ft. per in.	7.95 lbs.per sq.in.of L.P. piston per sq.in.	4.96 ft.-lbs. per sq.in. of L.P. area per sq.in.

If now total work is desired, the areas of the diagrams will be multiplied by the area scale and the area of each piston.

To refer these, however, to the L. P. area, the pressure scale for the H.P. cylinder will be multiplied by $\frac{A_{h.p.}}{A_{l.p}}$ and the I.P. scale by $\frac{A_{i.p.}}{A_{l.p.}}$. When this is done it is found that all area scales are the same.

$$31.25\times\frac{1}{6.3}=4.96;$$

$$8.99\times\frac{3.43}{6.30}=4.96.$$

Since then the tangential diagrams are of the same scale, they may be combined by addition. The customary arrangement of cranks as seen in Fig. 312 is to have them at 120° apart. Hence the three positions of $\theta=O°$ will be placed at 120° apart as in Fig. 283 and the resultant line found. The area of this resultant curve will be the energy put into the journal friction and if the height of the tangential effort required for this be laid off above the zero, it will be found that this will be the mean height of the curve and the amount of area above the line will be equal to the amount of area beneath the line.

These areas represent work or energy because the height represents tangential effort or force, while the abscissæ represent crank movement or motion in the direction of the force. To reduce these areas to actual total force they are multiplied by the area scale and the area of the low-pressure piston.

Calling the areas above the mean line excess areas, E_1, E_2, E_3, etc., and those below the line deficiency areas, D_1, D_2, D_3, etc., the variation of energy in the fly wheel may be found as follows:

At 1 the energy in the fly wheel may be called F; then by the time the crank has rotated to 2, the three systems have developed an excess of E_1 units of energy over the amount required by the pump. This energy must be absorbed by the fly wheel in an increase of speed. By the time 3 is reached, D_1 units have been abstracted, and so on for the various points; the energy at these is therefore as follows:

Point.	Energy.
1	F
2	$F+E_1$
3	$F+E_1-D_1$
4	$F+E_1-D_1+E_2$
1	$F+E_1-D_1+E_2-D_2=F$

If now the difference between the greatest and the least of these is found, that quantity, when multiplied by the area scale and the area of the low-pressure cylinder will give the amount of energy to be stored up in the fly wheel to change it from its lowest rate of speed during one revolution to its highest rate during that revolution. Calling this area difference ΔE and the moment of inertia of the fly wheel WR^2, where R is radius of gyration in feet and W the weight of the wheel and N′ and N″ the highest and lowest rates of speed in R. P. M. in one turn of the wheel, the following results:

$$\Delta E \times \text{area scale} \times \text{area } L\ P \text{ piston}$$

$$=\frac{WR^2}{2g}\left(\frac{\pi}{30}\right)^2[N'^2-N''^2];$$

now $$N'^2-N''^2=(N'+N'')\ (N'-N'')=2N\Delta N$$

where $$N=\text{mean speed}=\text{R. P. M.}$$

$$\Delta N=\text{variation in speed.}$$

The value of ΔN depends on the kind of machine considered. For slow pumps $\frac{1}{10}$ may be used while $\frac{1}{20}$ would be better for high speed pumps. (In the case of spinning mill engines and electric light engines the value $\frac{1}{100}$ is used.)

Assuming ΔN and the R.P.M. for a given pump, the quantity WR^2 is given by the formula. If R is known for a given shape, the weight W could be found.

In the case of the ordinary flywheel, the rim is the most important portion, and in many cases the rim is designed to give the necessary WR^2. In the case of the flywheel or rim, of width b, outer radius r_1, inner r_2; with hub, of width b^1,

outer radius r_3, inner radius r_4 and with n arms, the following results:

$$WR^2 \text{ for rim} = \int^{r_1} (w 2\pi r b dr) r^2$$

$$= \frac{\pi b w}{2} (r_1^4 - r_2^4)$$

$$= \pi b w (r_1^2 - r_2^2) \frac{r_2^2 + r_1^2}{2}$$

$$= W \frac{r_2^2 + r_1^2}{2}.$$

WR^2 for spoke, uniform elliptical section,

$$= n \int_{r_3}^{r_2} \frac{w \pi a c}{4} r^2 dr$$

$$= n \frac{w \pi a c}{4} \frac{r_2^3 - r_3^3}{3}$$

$$= W \frac{r_2^2 + r_2 r_3 + r_3^2}{3}.$$

$$WR^2 \text{ for hub} = W \frac{(r_3^2 + r_4^2)}{2}.$$

For whole wheel,

$$WR^2 = W_{\text{rim}} \frac{r_2^2 + r_1^2}{2} + W_{\text{spokes}} \frac{(r_2^2 + r_2 r_3 + r_3^2)}{3}$$

$$+ W_{\text{hub}} \frac{r_3^2 + r_4^2}{2}.$$

In this equation the last two terms may be found by taking data from practice, and then by assuming r_1 of the first term, the only unknown would be the term b of the weight W. If desired to assume b this may be done and then the unknown portion of the first term would be $r_1^4 - r_2^4$, in which r_2 is assumed, r_1^4 may be found.

If two flywheels are to be used, $\frac{1}{2} \Delta E$ is used for each wheel.

The first tangential diagram of Fig. 283 is the amount of twist which is transmitted from the first crank to the first fly-

wheel, while the third tangential diagram is the amount which is transmitted from the third crank to the second flywheel. The middle diagram is the amount which is transmitted from the center crank at the middle to the two wheels. This tangential effort, when multiplied by the crank radius, gives the twisting moment for which the shaft must be designed.

The net piston force from Fig. 281 gives the force which is applied to the crank shaft to produce bending. The amount in the direction of the piston is increased by the inclination of the connecting rod. The maximum increase amounts to only $1\frac{1}{2}$ per cent, and so this increase may be neglected in design, and the height from Fig. 281, when multiplied by the pressure scale and the area of the piston, will give the force which causes bending on the shaft. However this may be, after the pump is running, if anything should happen to the machine in one of the inner pump barrels, the plunger might jam and then the whole steam pressure from any one piston would reach the crank pin and hence the pin should be designed to support this.

CHAPTER VIII

STEAM END DETAILS

THE steam end of the pump has been illustrated in chapters III and IV quite fully, and it remains to examine peculiarities of design. The action of the steam end of simplex pumps has been fully studied and an explanation was made of the action of the valve of the duplex pump.

It was mentioned that in order to operate duplex pumps properly the steam valve should have a certain amount of play on the stem. This is accomplished in several ways. In Fig. 284

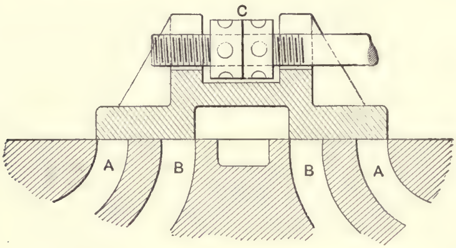


FIG. 284.—Steam Valve of Duplex Pump.

the method of using a central nut is shown. The valve is made without any lap, that is, the valve just reaches from the outer edge of one steam port *A* to the outer edge of the other, which would mean that as soon as steam was cut off from one end the other would open for steam; reverse the pump and so start the other pump, which in turn would reverse the front pump. By having the valve rod slide through slots in the projections on the back of the valve and by using the split nut of proper length to drive the valve, the action will be as follows: Suppose the valve is moved to the right by the motion of the other pump, the piston controlled by this valve will move to the right and it will start to move the valve rod on the other

pump to one side; however, the valve will not start until the piston, which is moving, reaches a point near the end of its stroke, whereupon the other piston starts and actuates the valve rod of Fig. 284 to move to the left. The valve does not operate until the rod has traveled the distance equal to the play between the nuts and the projection, at which time the valve will be moved to the left, closing the left-hand steam port and opening the right, thus starting the piston on its return. The time taken for one side to make the major part of its stroke is the time taken for the other side to come to rest at the end of its stroke and reverse. In this way there is a very steady discharge, as was explained earlier, since one piston is moving at full speed while the other is reversing. There is a small period of rest at the end of the stroke when the cylinder is full of steam, while the other piston is moving to throw over the valve. The driving piece *C* should be made in two parts so that they may be jammed to hold them at one place on the rod.

Another method of accomplishing this is by the use of outside striking pieces, as shown in Fig. 285. This arrangement is better, as the valve may be adjusted at either end. When this adjustment must be made often, it is done outside of the cylinder. One method is illustrated in Fig. 286. Here the valve stem *A* is provided with

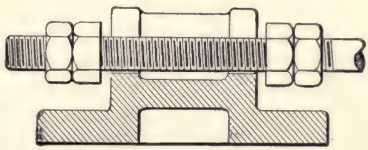


FIG. 285.—Steam Valve.

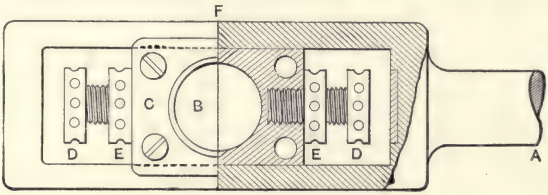


FIG. 286.—Valve Rod Yoke.

a box or yoke *F* on the end. This is driven through the cross head *C* by the pin *B* attached to the lever, connected to the other side of the pump. The amount of play is regulated

by the set screws DD which are fixed in place by the jam nut E.

In Fig. 287 another device is shown to accomplish the same result. The reverse lever A is driven from the piston rod B by a link. The lever is pivoted behind the valve rod. It moves

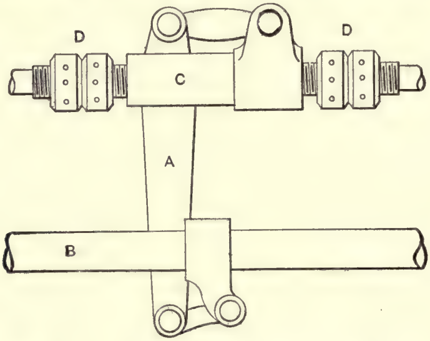


FIG. 287.—Driving Mechanism for Valves.

the sleeve C by means of a link and this sleeve strikes the jam nuts DD and moves the valve rod at the proper time.

These last two devices are used on large and small direct-acting pumps. They are used at times on the rotating steam valves which are the equivalent of D valves.

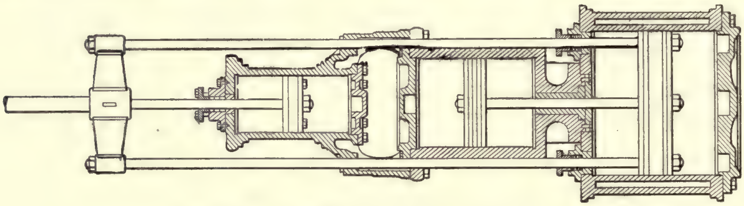


FIG. 288.—Triple Expansion Cylinders.

The cylinders of the single expansion duplex pumps are quite simple. The figures of chapters III and IV show their construction. The action of the five ports, the form of valve and piston, the arrangement of flanges for the cylinder heads, the valve chest covers, and the method of carrying the cylinders may be seen by reference to these figures.

The arrangement of the cylinders of a triple expansion direct-acting pump, Fig. 288, will serve to illustrate the methods of

building these. The cylinders are cast usually with single walls, and the cylinder ends are flanged to receive the cylinder head. Heads may be cast solid with the walls. This latter method serves to cut down the number of joints to be packed and does not materially increase the cost of construction. The heads of the cylinders are dished or cored to receive the nuts at the end of the piston.

The arrangement of piston rods shown in the figure is designed to reduce the number of inside packing boxes and make it possible to examine each piston without disturbing any cylinder.

When the cylinders are jacketed, several methods are

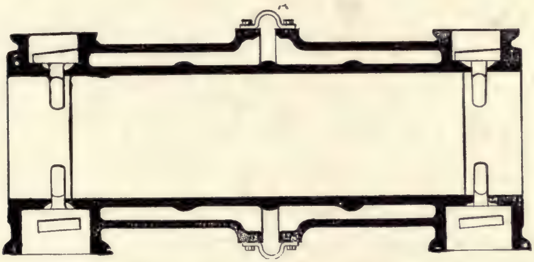


Fig. 289.—Leavitt Cylinder and Jacket.

available. The jacket may be made in the cylinder casting or it may be made by the use of a liner.

When the jacket is made in one piece with the cylinder barrel, a casting difficult to produce results, the outer shell is apt to cool first and when the inner part cools, strains are set up which crack the casting. To prevent this Mr. E. D. Leavitt, Jr., devised the scheme of casting the outer shell of the jacket with a division in the outer wall, Fig. 289. The opening thus made was closed by a copper ring *A*, with one corrugation. This ring was attached to the edges of the opening, by tap bolts with a sharp ring on the lower face of the head, which was driven into the copper and made a steam-tight joint. The edge of the copper was caulked against the cast iron.

The Snow Steam Pump Company makes jackets at times with a cast corrugation around the center of the shell to allow for this expansion and contraction. This cylinder,

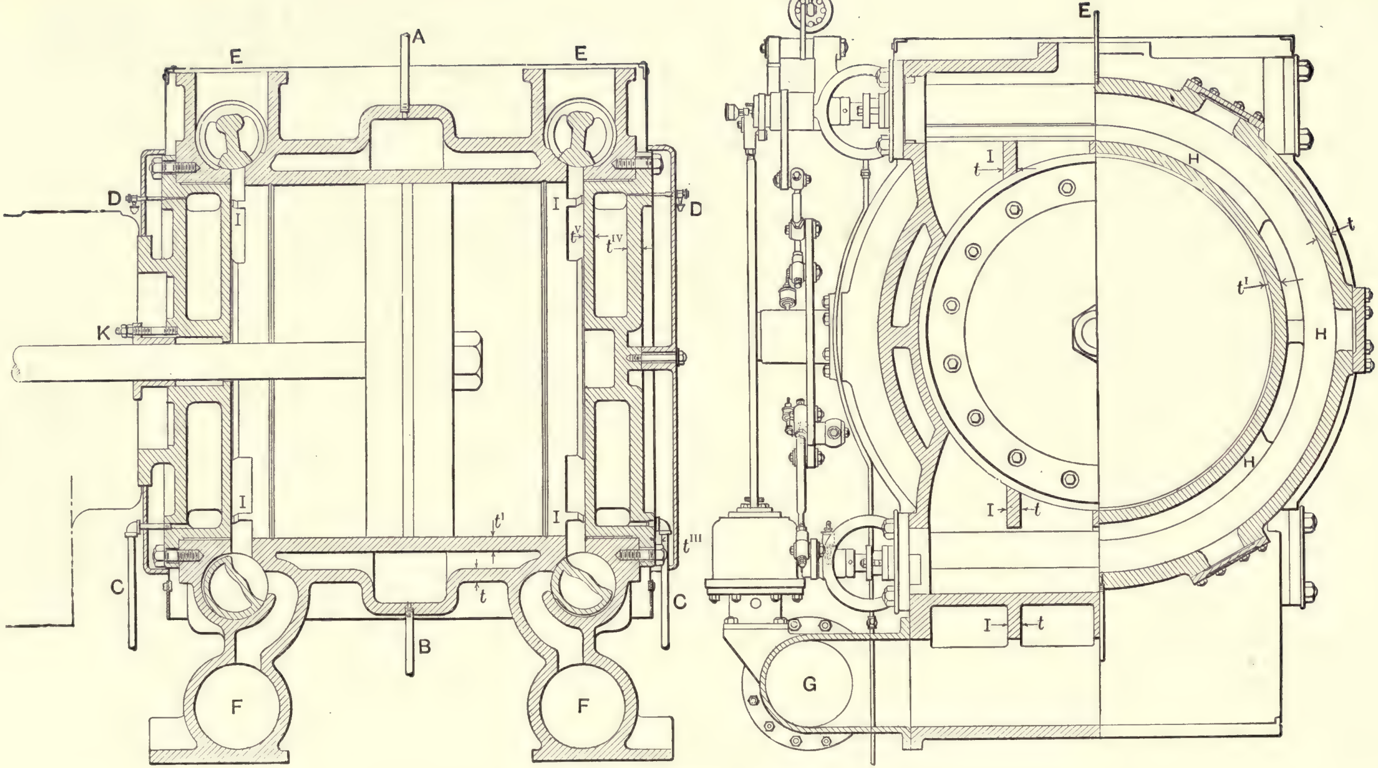


FIG. 290.—Steam Cylinder.

Fig. 290, is jacketed on the heads as well as on the barrel. The steam enters the barrel jacket at *A* and the condensation is removed at *B*. The condensation from the heads is removed by the drains *C*, which in some cases serve as admission pipes also, the system being the equivalent to a single pipe system of heating. The air valves at *DD* relieve the jacket of air.

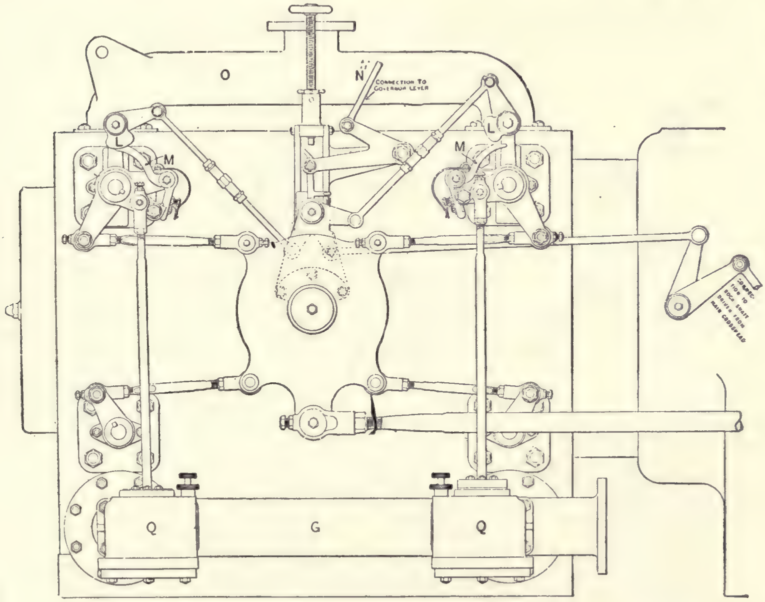


FIG. 291.—Valve Gear with Cut-off Controlled by Speed Governor.

The method of attaching the heads by stud bolts is clear. The stuffing box is shown at *K*.

The half cross sections through the valves and center, Fig. 290, show how the steam entering through *EE* passes the Corliss steam valves to the exhaust passages *FF*, which are cross connected to the manifold *G*.

The ribs at *H* and *I* are to stiffen the casting where passages are made.

The steam and exhaust valves and gear are shown in this figure and in Figs. 291, 292. In these the knock-off cam *L* and arm *M* are seen. In Fig. 291 the position of *L* is fixed by

the position of the governor through the rod *N* from the governor, and depends on the speed, while in Fig. 292 the position is controlled by the pressure in the force main from the pump.

The steam pipe *O* connects the inlets *E* leading to the valve chests, while the exhaust pipe *G* connects the exhaust outlets. The dash pots *QQ* are carried by the exhaust piping.

The jacket is sometimes constructed by introducing a liner within the cylinder, Fig. 293. In such design the liner

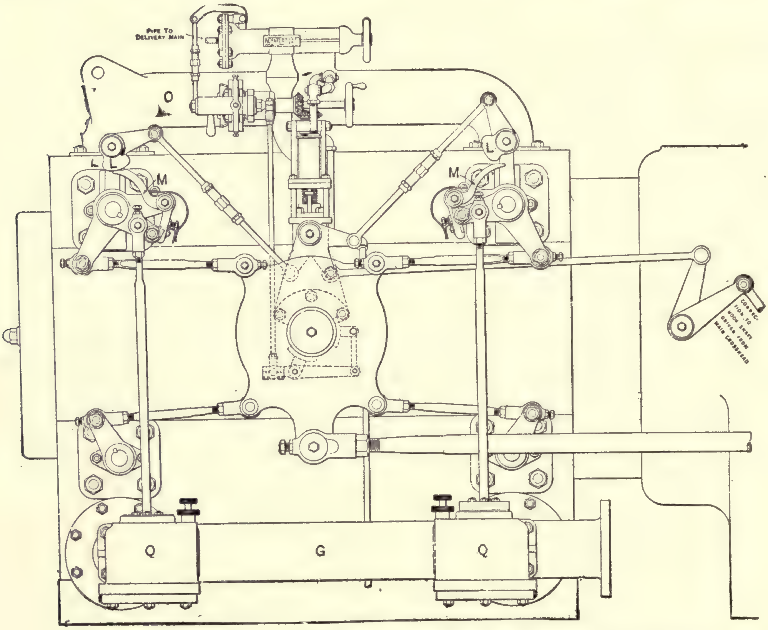


FIG. 292.—Valve Gear with Cut-off Controlled by Pressure Regulator.

may be made of a hard iron, while the main casting is of a soft cast iron, which is less liable to crack. The liner is fastened at its lower end by bolts, while at its upper end a copper ring is used, fastened by tap bolts with sharp circular lips on the lower faces. The jacket is supported by rings on its outer surface in contact with rings projecting from the inner surface of the shell. The surfaces of the projections are the only portions of the inner bore of the cylinder or the outer surface of the liner which require turning. Moreover, by making these of

decreasing diameter toward the closed end of the cylinder and by making each surface slightly tapered the liner may be introduced with little or no forcing and yet will be held tight.

The cylinder shown in Fig. 291 is intended for a horizontal

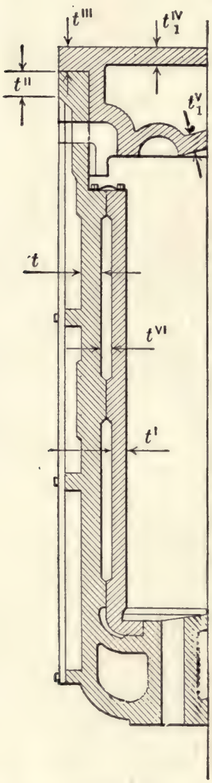


FIG. 293.—Independent Liner.

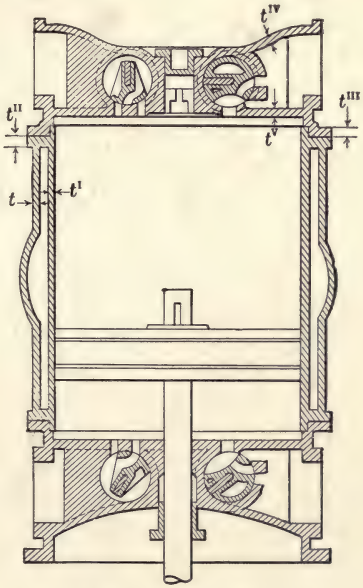


FIG. 294.—Head Valves.

engine, but the same construction would be used if for a vertical one. To cut down the clearance by eliminating the long passages extending from the straight valve face to the circular bore of the cylinder, engines have been built with the valves in the heads. Fig. 294 shows such an arrangement for Corliss valves. Many companies restrict this head valve to the low-pressure cylinders only. The system has the great advantage of simplify-

ing the cylinder casting, but for small high-pressure cylinders this is not so important.

The simple form of cored piston, Fig. 295, with two piston rings is a very common type. It is attached to the tapered end of the piston rod by a nut. A collar is used to keep the rod in position and support part of the thrust.

The half rings shown beside the piston illustrate the two forms of rings used with small pistons, one of constant cross section and the other with variable cross section.

Fig. 296 is the type used for large cylinders. The main

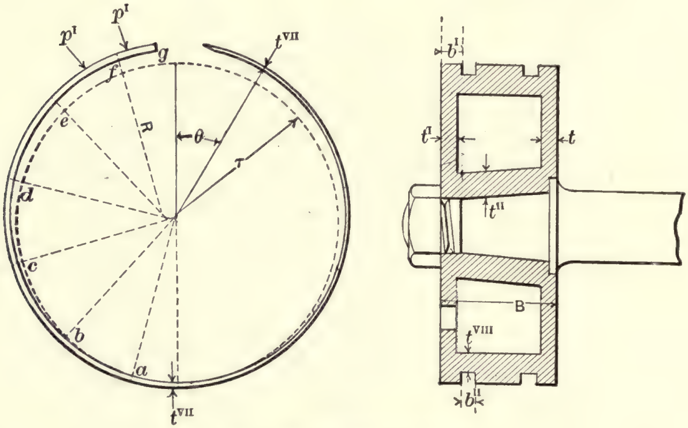


FIG. 295.—Steam Piston.

portion A is attached to the piston rod as shown, and from the ring B a number of bolts radiate, supporting a bull ring C. This bull ring contains a groove in which is placed a ring made of a series of overlapping sections. These are held out against the cylinder bore by helical springs in the cavities D. The follower plate E is held in position by the tap bolts FF. The ring B is supported by a series of webs from the central hub of the piston.

Fig. 297 illustrates another one of these pistons in which the sections are held out by carriage springs and the bull ring has two additional grooves which are filled with a bearing compound, so that the weight of the piston may be taken on

a substance which will not wear away the bore, but all wear will occur in the ring which may be turned as soon as the cast iron of the bull ring comes in contact with the bore. In this way

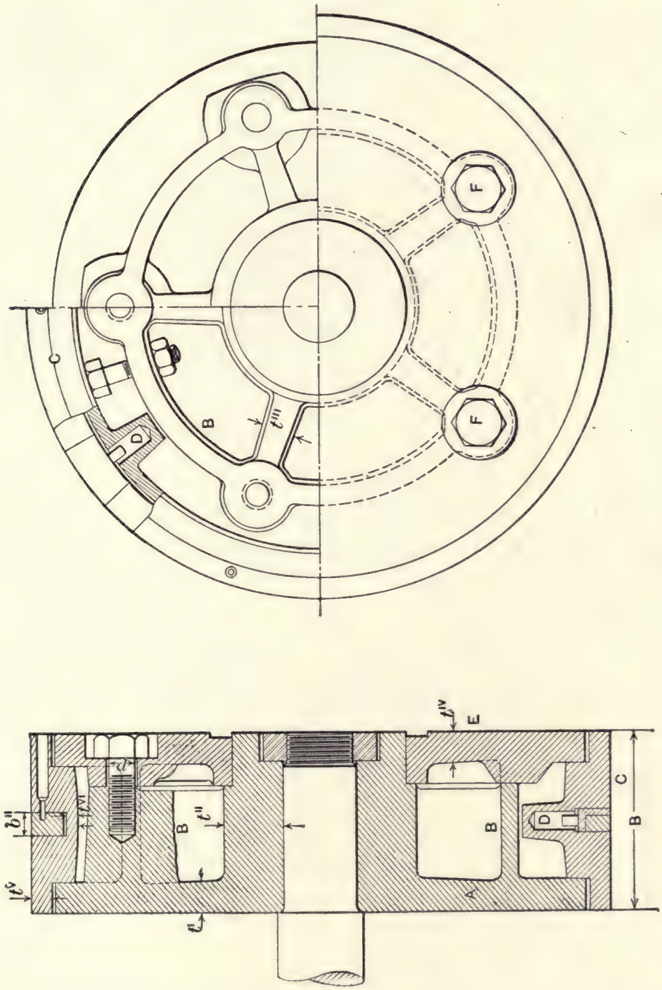


FIG. 296.—Piston.

these bearing rings will last for some time, as it will take eight or ten movements to use up the complete circumference. In Fig. 298 the bull ring is provided with two sectional rings.

These figures with those of the previous chapter illustrate the method of attaching pistons.

The simple stuffing boxes have been discussed under the

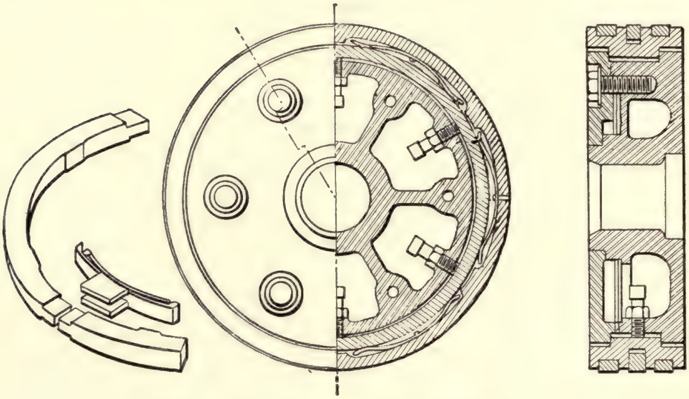


FIG. 297.—Piston with Sectional Ring.

head of water-end details. The metallic packing is quite common and as an example of one form Fig. 299 is given. In this the rings *A* are forced into the conical cups *BB* by the springs

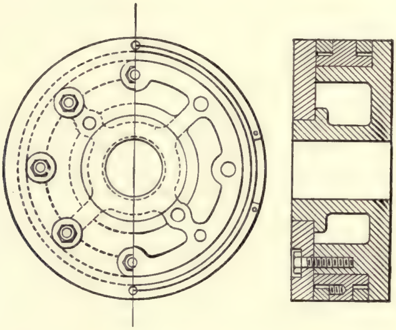


FIG. 298.—Piston.

C through the ring *D*. The cup *B* has a ground steam-tight joint against the ball-ended ring *E*. The ball *E* fits the socket *F*. The socket *F* can not be moved, the ball and socket allow for the alignment of the rod and the ground joint between *B* and

E allows the rod to move across the cylinder when wear of the piston or cross head makes this necessary. This is possible

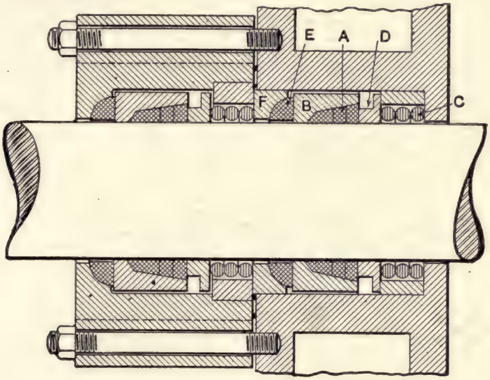


FIG. 299.—Metallic Packing.

since the rings or cups E and F are all of a larger inside diameter than that of the rod.

DESIGN OF CYLINDER

The thickness of the cylinder is given by the same formula as suggested in chapter VI. (See Figs. 290, 293 and 294.)

$$t=\frac{dp}{5000}+\tfrac{1}{2},$$
$$t'=0.8t+\tfrac{1}{4}'',$$
$$t''=1.25(t+\tfrac{1}{4}),$$
$$t'''=1.25(t+\tfrac{1}{4}),$$
$$t_1^{\text{v}}=t_1^{\text{iv}}=0.6 \text{ thickness of single plate cover.}$$

Thickness of single plate cover $=0.55d\sqrt{\frac{p}{s}}$,

$$t^{\text{vi}}=\tfrac{1}{2}'' \text{ to } 1'' \text{ for liner jackets.}$$
$$=1'' \text{ to } 3'' \text{ for cast jackets.}$$

The valve chest walls and covers have their thicknesses determined by the cylinder thickness formula or the flat surface formula.

$$t^{\text{vii}}=0.35d\sqrt{\frac{p}{s}}\left(\text{or } 0.55d\sqrt{\frac{p}{s}} \text{ for round plates}\right).$$

Flanges here are made $1.25t^{vii}$ in thickness.

The diameter of the bolt should vary with the size of cylinders. $\frac{5}{8}''$ bolts are the smallest to be used and these may be increased to $1\frac{1}{2}''$ on very large cylinders. $\frac{3}{4}''$ and $1''$ bolts are common on cylinders of 20″. After the diameter is assumed the number of bolts is found by the formula

$$p\frac{\pi d^2}{4}=nS_t a;$$

a = area at root of thread of bolt in sq.in. from table of standard threads.

S_t = allowable stress in pounds per sq.in.

= 4000 lbs. per sq.in. to allow for straining.

n = number of bolts.

p = maximum steam pressure in lbs. per sq.in.

d = diameter of cylinder in inches.

Flange width $3d$ at least.

Pitch of bolts $< 40\sqrt{\frac{d}{p}}$.

Steam Passages

The area of steam passages is given by

$$a=\frac{2LNA}{6000},$$

while the exhaust passages are given by

$$a'=\frac{2LAN}{4500}.$$

In these a is the area of cross section of tne passage in square inches; L, the stroke in feet; N, the revolutions per minute, and A, the area of the piston in square inches.

The length of the port is made about 0.8 diameter of cylinder, then, width $=\frac{a}{0.8d}$. The value 6000 for the velocity of steam should only be used to find the steam passages. The port-opening is found by using this value when cut-off is at 0.7

stroke. For 0.25 stroke, the port-opening is found by using 18,000 for the velocity. For cut-offs between these use a proportional amount.

Clearance

The clearance distance is used to allow for wear in the connecting rod and bearings, unevenness in the castings of the piston and head, and errors in alignment. The clearance distance is made from $\frac{3}{16}''$ to $\frac{9}{16}''$ in different sizes of engines, the first being used for strokes of 15″ and under. The latter for strokes to 6 feet.

The clearance volume is given by

$$cl = AK + aL_1 + a'L_1',$$

$$\% \ cl = \frac{AK + aL_1 + a'L_1'}{LA}.$$

In these L_1 and L_1' are the length of the steam and exhaust passages.

The clearance has an important effect, as has been shown in many engine tests, so that it is desired to keep the value of this as small as possible. The following table will give some idea of the variation of this quantity.

	Clearance Volume in Percentage of Cylinder Displacement.		
Simple Engine:			
D-Slide valve	6 to 15		
4 valve engines	2 to 4		
Corliss valves	2 to 4		
Piston valves	7 to 15		
Compound:	H.P.		L.P.
Slide valve	7 to 15		5 to 10
Corliss valves in side	2 to 4		1.3 to 3
Triple:	H.P.	I.P.	L.P.
Slide valves	10–15	5–10	5–7
Corliss valves	1.5–1.75	1.25–2.0	1.5–2.5
Corliss valves in head	——	$\frac{1}{2}$–1	$\frac{1}{2}$–1

PISTON PROPORTIONS

(See Figs. 295 and 296.)

The design of the piston is empirical and the following proportions are recommended by various authors:

$$\text{Flat disc } t = \frac{D\sqrt{p}}{100} \text{ for cast iron.}$$

$$= \frac{D\sqrt{p}}{150} \text{ for steel.}$$

This value will be used as the unit for all pistons; and dimensions on the figures are in terms of this:

B $=0.4D$ and for large cylinders $\sqrt[4]{LD}$ (inches).
Thickness of faces $=t'=0.5t$.
Thickness of face when follower ring is used $=0.8t$
Thickness of center boss $=t''=0.5t+t'''$.
Thickness of ribs $=t'''=0.3t$
Distance from edge to ring, $b'=0.5t$.
Thickness of junk ring or follower plate $t^{iv}=0.5t$.
Diameter of junk ring bolts $d'=0.25\ t$.
Number of webs $=0.1D+2$.
Thickness of bull ring at edge $t^{v}=0.6t$.
Thickness of bull ring under packing ring $t^{vi}=.5t$.
Thickness of spring packing rings $t^{vii}=0.03D$.
Width of spring packing rings $b''=\frac{1}{2}''$ to $\frac{3}{4}''$.
Thickness of solid piston wall beneath ring $t^{viii}=t$.
Diameter of piston ring $=1.015D$.

The basis for the proportions used for the piston rings above as given by Unwin in his "Machine Design," is as follows:

Assume the piston ring, Fig. 295, to be loaded with p' pounds pressure per square inch and to be changed by this from a radius R at any point to the radius r of the cylinder. If the width of the ring is b'' the resultant pressure causing bending at the angle of θ from the end after fitting into the cylinder is

$$P=(2r \sin \tfrac{1}{2}\theta)b''p',$$

and its moment is

$$M=\left(2r\sin\frac{\theta}{2}\right)b''p'r\sin\frac{\theta}{2}$$

$$=2p'b''r^2\sin^2\frac{\theta}{2}.$$

This bending moment is equal to $\frac{SI}{C}$ or

$$M=\frac{Sb''(t_x^{\text{vii}})^2}{6}=2p'b''r^2\sin^2\frac{\theta}{2}.$$

The change in curvature also depends on this moment or

$$\frac{1}{r}-\frac{1}{R}=\frac{M}{EI}.$$

This last expression is used to determine the variation in curvature when I is constant, or the variation of I if the curvature before springing into the cylinder is constant. The first equation is used to determine the thickness at the point in the ring half way from the ends. In this case $\theta=180°$ and

$$M=2p'b''r^2=\frac{Sb''(t^{\text{vii}})^2}{6}.$$

Using 6000 for S and 2 pounds for p' the following results:

$$2\times2\times r^2=\frac{6000t^2}{6};$$

$$t=r\sqrt{\frac{24}{6000}}=.06r.$$

If now the section is made uniform, as on the left of Fig. 295, the value of R will have to be different at various points.

$$I=\frac{b''(0.06r)^3}{12}=0.000018b''r^3.$$

$$\frac{1}{R}=\frac{1}{r}-\frac{2p'b''r^2\sin^2\frac{\theta}{2}}{E\times0.000018b''r^3};$$

for $$p' = 2 \text{ lbs. and } E = 15{,}000{,}000.$$

$$\frac{1}{R} = \frac{1}{r} - \frac{4r^2 \sin^2 \frac{\theta}{2}}{270r^3};$$

$$R = \frac{270r^3}{270r^2 - 4r^2 \sin^2 \frac{\theta}{2}} = \frac{270r}{270 - 4 \sin^2 \frac{\theta}{2}}.$$

For different values of θ the values of R are given below:

$\theta = 0°$	$30°$	$60°$	$90°$	$120°$	$150°$	$180°$
$R = 1.000r$	$1.003r$	$1.004r$	$1.007r$	$1.011r$	$1.014r$	$1.015r$.

The curve of the original curvatures is drawn in Fig. 295; the radius for 180° being used for a length of arc equal to 15° of the cylinder circumference on each side of the 180° position. This gives the point a, and on the radius to this point a new center is taken equal to that for $\theta = 150°$. This arc is made equal to 30° of the cylinder circumference, giving the point b. On the radius to this point a new center is taken for the curve to c, then in a similar manner d, e, f and g are found. When this ring is sprung together until the points touch, it will just fit in the cylinder and will exert a uniform pressure of 2 pounds per square inch.

If R is constant the expression for curvature becomes

$$\frac{1}{r} - \frac{1}{R} = K = \frac{4b''r^2 \sin^2 \frac{\theta}{2}}{Eb''\frac{(t^{vii})^3}{12}};$$

or $$t^{vii\,3} = Kr^3 \sin^2 \frac{\theta}{2};$$

$$\frac{t_\theta^{vii}}{t_{180}^{vii}} = \frac{\sin^{\frac{2}{3}} \frac{\theta}{2}}{1};$$

$$\therefore \quad t_\theta^{vii} = t_{180}^{vii} \sin^{\frac{2}{3}} \frac{\theta}{2}.$$

Hence the following results in terms of t_{180}^{vii}:

$\theta=0$	5	10	15	30	45	60
$t^{vii}=0$	.124	.197	.257	.405	.529	.630

90	120	150	180
.794	.908	.971	1

The thickness t_{180}^{vii} is made the same in this as in the former method.

Piston Rod

The piston rod is designed as described in chapter VI, page 314.

For the remainder of this chapter the maximum pressure on the piston will be called P, where

$$P=\frac{\pi d^2}{4}p;$$

$$A_t=\frac{P}{S_t};$$

$$A=\frac{P\left(1+K\frac{l^2}{r^2}\right)}{S_c};$$

A_t = area at root of thread;
A = area of main rod.

Shoulder $\frac{1}{8}''$ to $\frac{1}{4}''$ with taper of $3''$ in $12''$.

Stuffing Box

The stuffing-box design has been given in chapter VI, page 201.

Cross Heads and Connecting Rods

The cross head is necessary to guide the piston rod and keep it from deflecting under compression and also to take the side thrust from the connecting rod. The amount of this side thrust

is shown by a force diagram in Fig. 300. The three forces, P from the piston rod, P' from the connecting rod and R from the guides are in equilibrium. R and P' will be the largest for a given value of P when α is a maximum. Sin $\alpha=\frac{ab}{ac}$. ac is constant. Hence sin α is a maximum when ab is a maximum. This occurs when $ab=oa$ or θ is 90°. In this case the length of

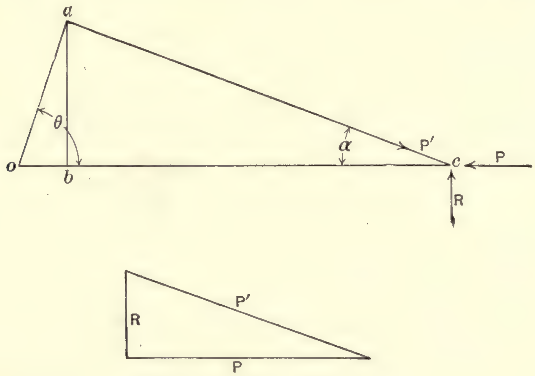


FIG. 300.—Force Diagram.

P' in force polygon is n times the length of R and the following relations hold:

$$P'=\frac{n}{\sqrt{n^2-1}}P,$$

$$R=\frac{1}{n}P'=\frac{1}{\sqrt{n^2-1}}P.$$

(n = ratio of connecting rod length to crank arm.)

Although P rarely continues its full value until the crank has moved 90°, since cut off occurs earlier than this, it is well in design to consider this to happen and design accordingly. This load might be developed if the full pressure were continued.

The form of cross-head used in pumps when the water end is in tandem with the head end of the steam cylinder or where the pump is driven from another part of the shaft, as in a triplex pump, is shown in Fig. 301.

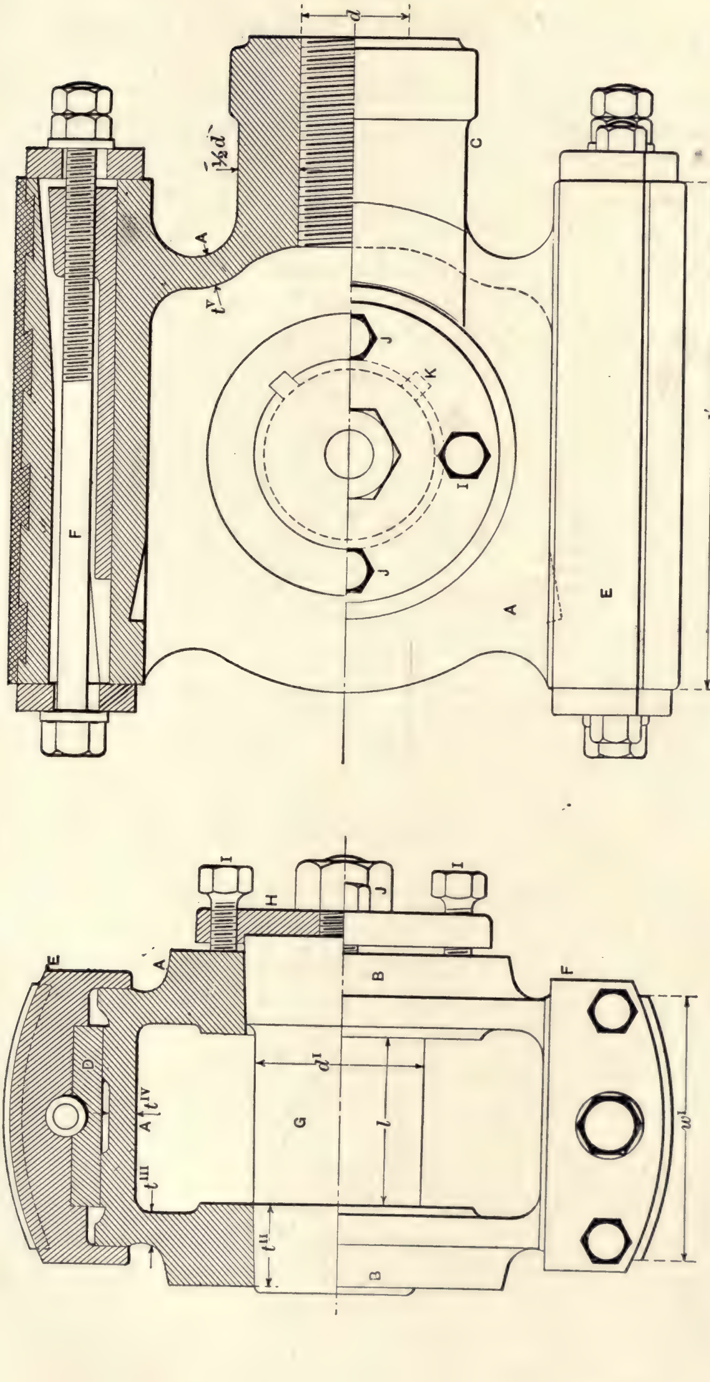


FIG. 301.—Cross-head.

This type is not the principal type used, but it is of good form and is therefore given.

The main casting A is made like a rectangular box with one end open and bosses BB placed on two sides, while a longer boss or tube C is placed on the side opposite the open end. On the upper and lower sides of the main casting A are placed slippers or shoes EE which are supported by wedges D. The lower face of the wedge D rests on the horizontal surface of A, but its upper face is inclined and supports the inclined face of E. The wedge may be moved by turning the bolt F, which is jammed after moving the wedge far enough to lift the slipper out from

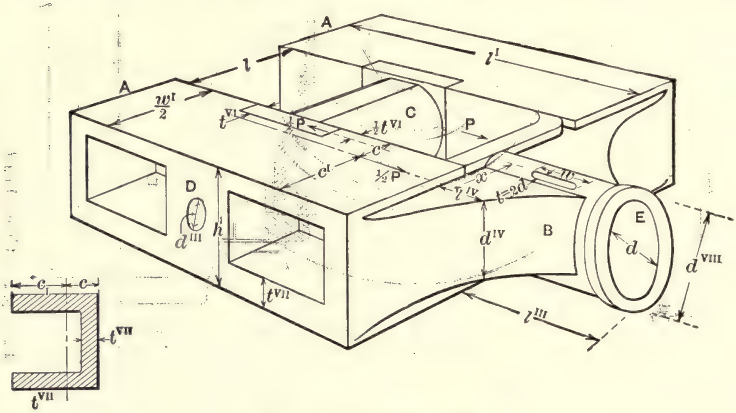


FIG. 302.—Cross-head.

the casting an amount equal to the wear. The slippers are faced with a bearing metal such as babbit.

The pin G is slightly tapered where it passes through the sides of A. This makes it possible to have the pin tight without driving. The pin is fastened by a nut to the plate H which is held in position by two tap bolts JJ while the set screws II jam the plate so that there is no danger of the bolts JJ backing off.

The piston rod is screwed into the boss C and a jam nut keeps it from backing off.

The cross-head, Fig. 302, illustrates a form which is used when four guide bars are employed. The main casting consists

of two blocks AA connected by the yoke B. The cross-head

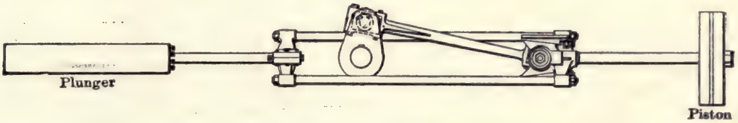


FIG. 303.—Crank and Reciprocating Parts.

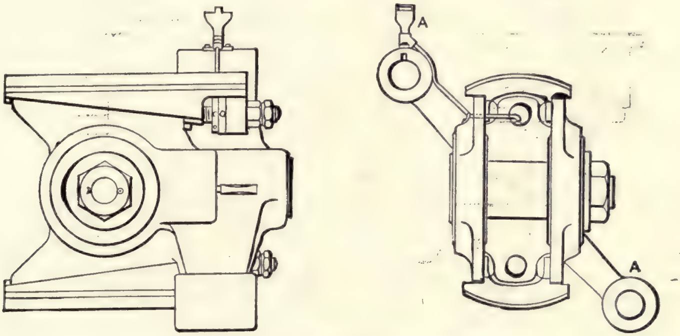


FIG. 304.—Two-rod Cross-head.

pin C is separate from the casting and is held in position in the grooves made for it by the pin D. The piston rod fits into the tapered hole at E and is held in place by a cotter.

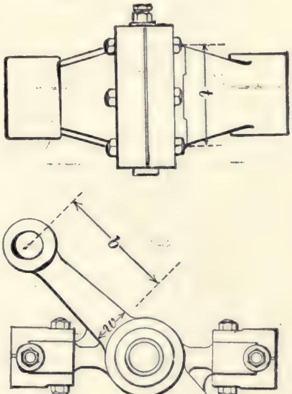


FIG. 305.—Pump End Cross-head.

Fig. 303 shows the arrangement of the steam piston and water plunger when these are in tandem with the crank-shaft between them. When this is the case the cross-head has to be so arranged that the rods which are used to cross over the crank and shaft may be fastened to the cross-head. As shown, this is accomplished by two rods, one above and behind the center line of the piston rod and the other below and in front so fixed that they are symmetrical about the center line.

The cross-head at the steam end is shown in Fig. 304. It is

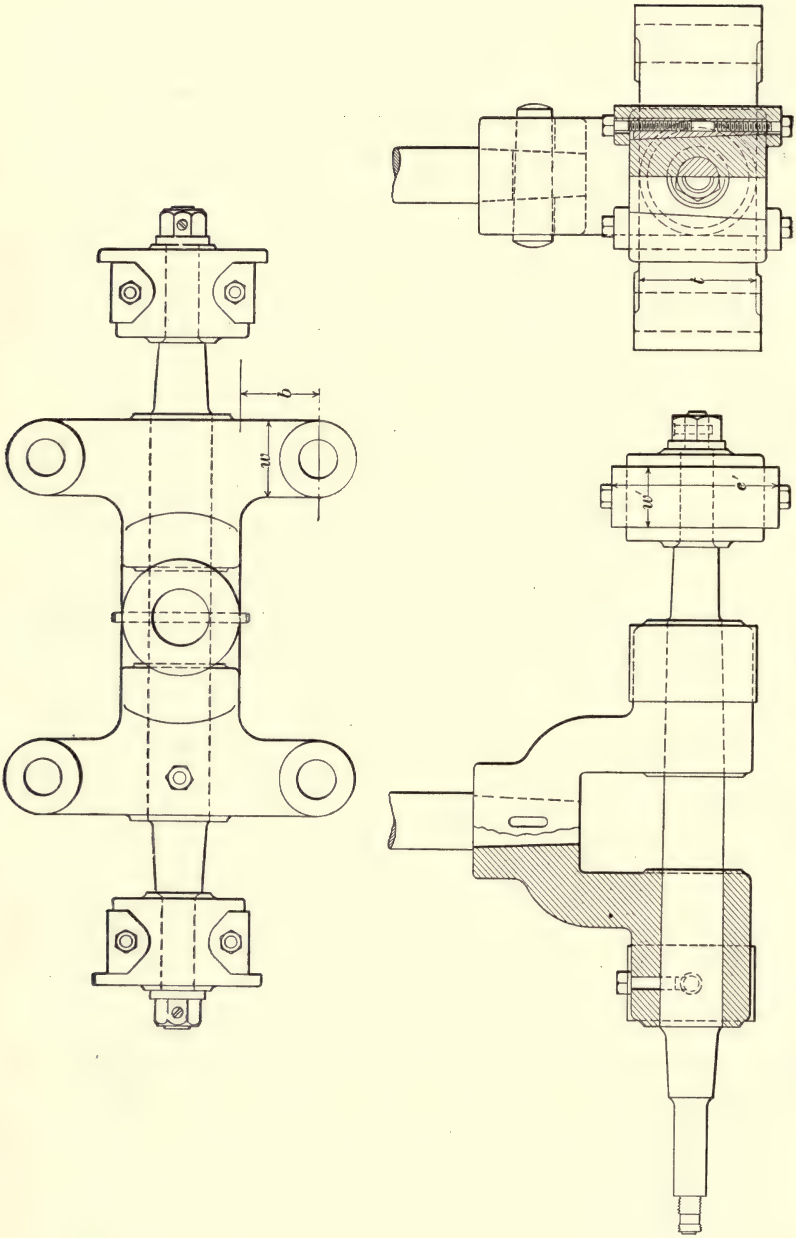


FIG. 306.—Four-rod Cross-head.

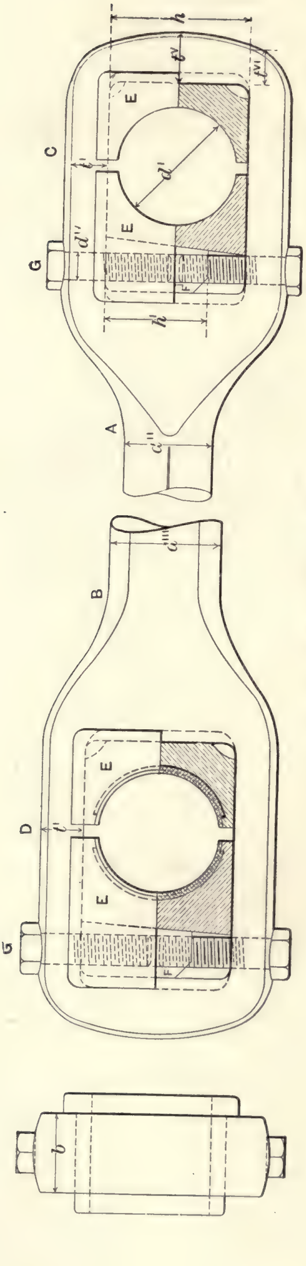


FIG. 307.—Connecting Rod.

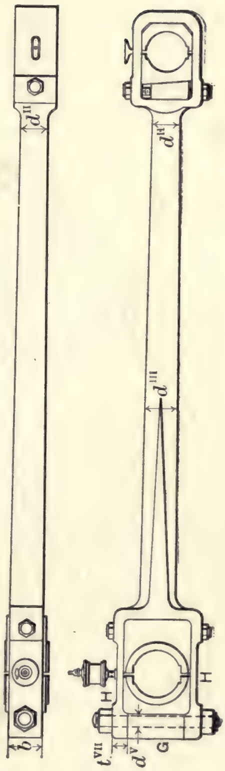


FIG. 308.—Connecting Rod.

of the same form as Fig. 301 with the addition of the arms *AA*. The water end cross-head, Fig. 305, is only used to prevent buckling of the rods when under compression and hence is not so large as the other.

In some cases the designer prefers to use four rods symmetrically placed, and in such a case the form used is that shown in Fig. 306. This form is modified on account of the great distance between the frames in which the guide surfaces

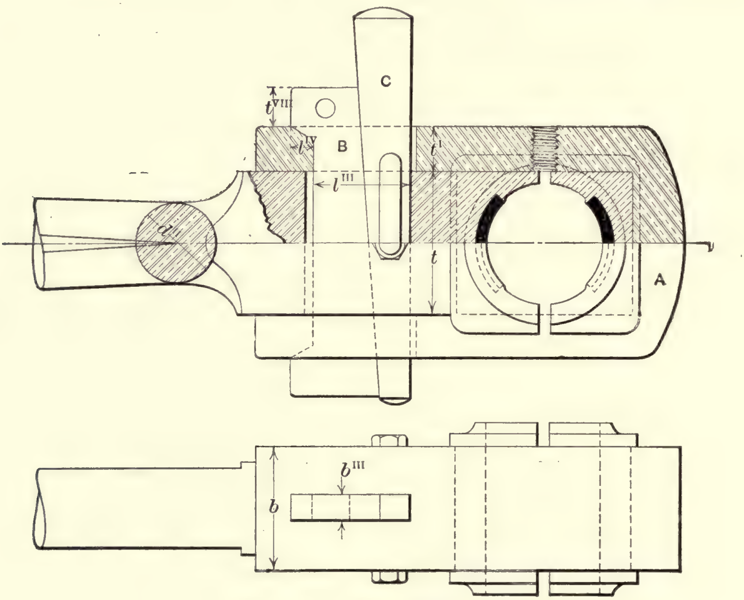


FIG. 309.—Strap End.

must be placed. The sliding blocks, as seen in the figure, are cast blocks fitted to the pin at the center, which passes through the main portion and forms the wrist pin. These blocks are used for guiding only, the largest part of the steam forces going directly from the piston rod to the pump rods.

The types of connecting rods used on these machines are shown in Figs. 307 and 308. In Fig. 307 the main forging is made with boxes at each end and is circular in section at *A*, being turned from *A* to *B* as a cone with its faces slabbed off. The box ends *C* and *D* are fitted with brasses *EE* which are

tightened up for wear by the wedges FF. The wedges are moved by the bolts G which are jammed by the nut on their opposite faces. The wedges are arranged to shorten one end, as wear occurs while the other end is lengthened.

In Fig. 308 the rod is made with a conical section in which the slabbing of the sides occurs near the crank end. In this

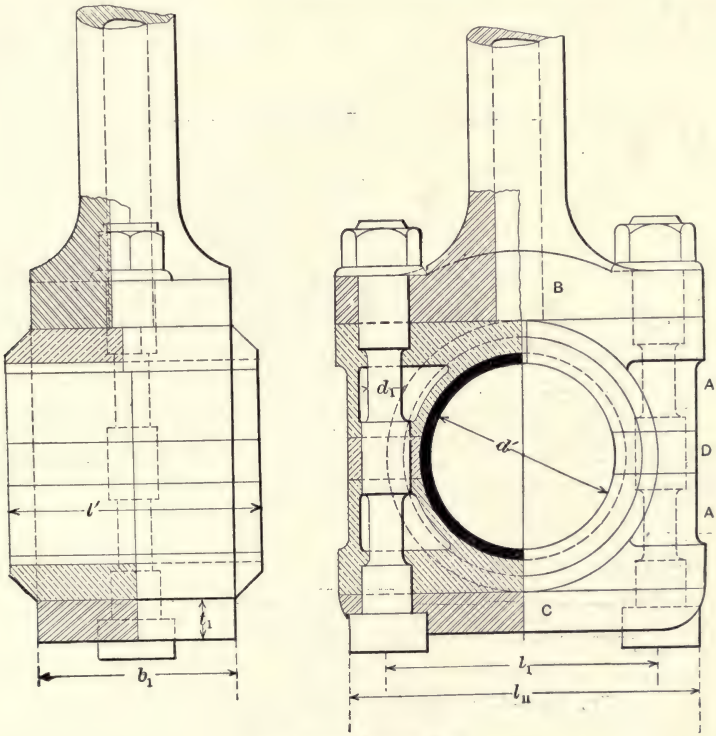


FIG. 310.—Marine End.

rod the box at the crank end is made by inserting a block G between two fork ends HH and bolting it. This is necessary when the rod is used with the center crank A of Fig. 312. The rod 307 could only be used with the overhung crank B of Figs. 311 and 312.

The strap end 309 and the marine end 310 may be used with center cranks. In the strap end, Fig. 309, the strap A is held on the square end of the rod by the gib B and the key or cotter

C inserted in a slot through the end of the rod. The key is prevented from coming out of the slot by a set screw, and the end of this set screw engages with the key in a groove so that the burr formed does not interfere with the removal or adjustment of the key. The brasses and other parts are the same as in the previous rods. The marine end, Fig. 310, has two brasses

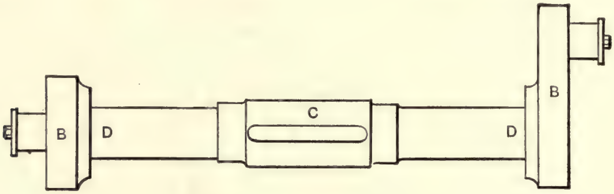


FIG. 311.—Crank Shaft.

AA held to the club end *B* of the rod by a keeper *C* and two bolts. The liners *D* allow the bolts to be made tight and jammed without gripping the crank pin. When wear occurs these are reduced in thickness.

The crank-shaft for a two cylinder engine is shown in Fig. 311, while in Fig. 312 a three-crank shaft is illustrated. In these the arms *BB* may be driven on and keyed or in some cases they

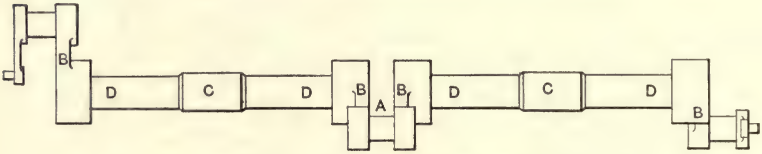


FIG. 312.—Three-crank Pin Shaft with Return Cranks.

are forged solid in one piece. The bearing journals are at *DD* and at times the shaft is enlarged at the point of attachment of the flywheel, as at *C*. A key-way is left for the flywheel. These shafts are sometimes forged hollow to reduce weight and to remove metal which is of little value.

DESIGN OF CROSS-HEAD, CONNECTING ROD AND SHAFT

The designs of the cross-head, connecting rod and shaft are interdependent so that it is well to take up the designs together.

THE CROSS-HEAD

The pressure on the cross-head to be carried from the piston is P pounds, P' pounds from the connecting rod and R from the guides, where

$$P = \frac{\pi d^2}{4} p,$$

$$P' = \frac{n}{\sqrt{n^2 - 1}} P,$$

$$R = \frac{1}{n} P'.$$

If the connection with the piston rod is made by a thread, the amount of thread to be covered by the boss of the cross-head should be equal to the diameter of the rod. If a key is used to key the rod to the cross-head the thickness of the key (t, Fig. 302) should be $0.2d$ and the width is given by

$$w = \frac{P}{2 \times 0.2 d S_s}.$$

The pin should be investigated for crushing and the rod for bearing by the formulæ:

$$0.2d \times dS_c > P$$

$$\left(\frac{\pi d^2}{4} - 0.2d^2\right) S_t > P.$$

The cross-head pin may be designed empirically and then investigated for bearing, bending, and deflection. A good rule is to make the length of the pin (l – Figs. 301 and 302) equal to $\frac{\sqrt{P'}}{30}$, and the diameter (d' – Fig. 301 and 302) $\frac{\sqrt{P'}}{40}$.

The bearing pressure $\frac{P'}{ld'}$ then becomes 1200 pounds per square inch, which is allowable for wrist pins.

The investigations for stress and deflection are given by the formulæ:

$$S_s = \frac{P'}{2\frac{\pi d'^2}{4}},$$

$$M = \frac{P'l}{8} = < S_t \frac{\pi d^3}{32}.$$

$$\Delta = \frac{1}{100} = > \frac{5P'l^3}{384E\frac{\pi d^4}{64}}.$$

These are usually fulfilled, but it is well for the student to investigate expirical design.

The area of the shoe is designed so as to give a pressure of 40 pounds per square inch in fast-running engines with cast iron slippers and 250 pounds in slow engines when the shoe is lined with babbit metal and 80 pounds in case of cast-iron slippers against cast-iron guides for slow running.

$$A' = l'w' = \frac{R}{p'}.$$

p' is the allowable pressure for any given problem.

The relative values of l' and w' are fixed by the design.

$\frac{1}{4}l' = w'$ may be used.

The pin is held by the sides of the main casting. In Fig. 301 the supporting surface is $2d't''$.

$$t'' = > \frac{P'}{2d'S_c}.$$

The details of the main casting (Fig. 301) are given in terms of d'.

Thickness side, $t''' = 0.3d'$, but must be not less than $\frac{1}{2}''$,
Thickness boss, $t'' = t''' + \frac{1}{2}''$,
Thickness bottom, $t^{iv} = 0.25\ d'$,
Thickness front, $t^{v} = 0.25d'$.

In the cross-head shown in Fig. 302 the design is somewhat different. Here the thickness must be calculated.

$$h' = d' + \tfrac{1}{2}'',$$

$$t^{vi} = \frac{P}{2S_c h'},$$

$$d''' = \sqrt{\frac{P'}{2S_s \frac{\pi}{4}}}.$$

The cross-head pin is cut off for 20° at top and bottom as this is not of much value and it distributes the oil better when so treated.

The thickness of the side blocks t^{vii} is assumed and then investigated for strength. The piece is under combined bending and direct stress and is investigated as follows:

Fig. 302 shows the general arrangement of the forces and their positions. The thickness t^{vii} may be assumed and then the center of gravity c, moment of inertia I, and area of section A', shown at side, are found. The point of application of the load $\frac{P}{2}$ is at $\frac{1}{2}t^{vi}$ from the edge so that there is a lever arm of $c - \frac{1}{2}t^{vi}$ causing bending. On the crank-head stroke the maximum tension is

$$S_t = \frac{P}{2A'} + \frac{P(c - \frac{1}{2}t^{vi})c}{2I};$$

on the other stroke the maximum compression is the same. Since the tensile strength is usually less than that of the compressive strength for the material used for cross-heads, the investigation for S_t is the only one made. If S_t is found to be greater than the allowable S_t, the section must be increased or cast-steel used in place of cast-iron.

The yoke is designed as a cantilever beam. At the center it is best to assume d^{viii} and compute l''' by the formula

$$\frac{P}{2} \times (\tfrac{1}{2}l + c) = \frac{S_t(d^{viii} - d)l'''^2}{6}.$$

For a section at x from the center, the thickness d^{iv} is assumed and the distance l^{iv} is found by the formula:

$$\frac{P'}{2}\left(\tfrac{1}{2}l+c-x\right)=\frac{S_t(d^{iv})l^{iv2}}{6}.$$

In Fig. 305 the arms are designed as cantilever beams by assuming t and finding w from the formula:

$$\tfrac{1}{2}Pb=\frac{S_t wt^2}{6}.$$

In Fig. 306 the bending moment is $\frac{1}{4}Pb$. Although the force P is in general not the whole force on the steam piston, yet the design should be made to carry this load since that force might come upon it.

The length of the crank pin is determined by the heating caused by friction. The total piston pressure P will produce the amount of friction μP where μ is the coefficient of friction. This force moves through the distance $rd\theta$. The work per minute will be

$$W=N\int_0^{2\pi}\mu Prd\theta=\mu Nr\int Pd\theta.$$

If this is divided by the product of the length and the diameter of the pin $l'd'$, the result will be the amount of work per square inch of projected area of pin. This quantity then becomes

$$w=\frac{W}{l'd'}=\frac{\mu N}{2l}\int_0^{2\pi}Pd\theta$$

If for $\int_0^{2\pi}Pd\theta$ the approximate value, $2\pi P_{\text{mean}}$ is used, the above expression becomes

$$w=\frac{\pi\mu NP'_{\text{mean}}}{l'}.$$

By using the values of μ, N, P and l in a number of engines which have run without heating, a mean value of w may be found and from this value on substitution a formula for l may be derived:

$$l'=K\mu P_{\text{mean}}N,$$

since
$$P_{\text{mean}}=\frac{\text{I.H.P.}\times 33{,}000}{2LN},$$

$$l'=K'\frac{\mu\text{I.H.P.}}{L}.$$

These two formulæ may be used to determine l' by making $K=0.00003$ and $K'=0.7$ to 1.0.

The value of μ is 0.04 for proper lubrication and 0.10 for poor lubrication.

After the length of the crank pin is determined its diameter is found to care for strength, bearing pressure and deflection. These three are considered separately. If the crank pin is overhung, the pin acts as a cantilever beam. Then

$$\frac{Pl'}{2}=S_t\frac{\pi d'^3}{32},$$

$$d'=2\sqrt[3]{\frac{2P_1l'}{\pi S_t}}.$$

For bearing
$$p=\frac{P}{l'd'}$$

or
$$d'=\frac{P}{l'p}.$$

In this p is the unit bearing pressure. For slow engines Unwin allows 800 to 900 pounds per square inch. While on fast engines the value is from 500 to 800, in marine engines 400 to 500. For deflection the formula is

$$\Delta=\tfrac{1}{8}\frac{Pl^3}{EI}=\frac{8Pl^3}{E\pi d^4},$$

$$d=\sqrt[4]{\frac{8}{\pi}\frac{Pl^3}{\Delta E}}.$$

The amount of deflection is limited to 0.01 inch. If the crank pin is a center pin, the design for strength is to be considered with the design of the shaft, and for that reason the shaft design will now be studied.

In beginning to design a shaft, assumptions must be made

of the various lengths. The length of the pin l' has been found and this will serve as a guide for certain other parts. It may be assumed that crank arms have the width l' and that the bearings have a length $2l'$. With these dimensions in view an approximate sketch is made. Fig. 313 shows this for an

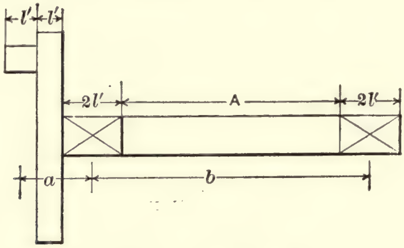


FIG. 313.—Shaft Sketch.

overhung crank and Fig. 314 for a triple crank-shaft, both of approximate form.

The distances A are assumed of sufficient size for the fly-wheel or to bring the cylinders of the multicylinder engine at the proper distances apart. The quantities, a, b, c, d, e, etc., are now found in actual lengths in inches.

From the diagrams of Fig. 281 the greatest steam pressure

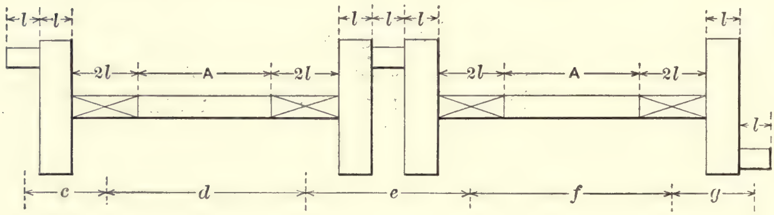


FIG. 314.—Shaft Sketch.

can be found and the crank position at which it acts can be constructed, giving a diagram as Fig. 315 from which P' is found. This force acts on the crank pin, but its equivalent P' acts at the center of the shaft in connection with a couple. This is seen in the figure by adding two equal and opposite forces at the center of the shaft. One of these combines with P' to form a couple which produces twisting only while the

other force left at the center of the shaft produces bending only. The other piston forces on a triple crank-shaft may be found by obtaining the positions of the other pistons from a diagram similar to Fig. 282, and then from the piston position obtaining the pressures from Fig. 281 by multiplying the heights

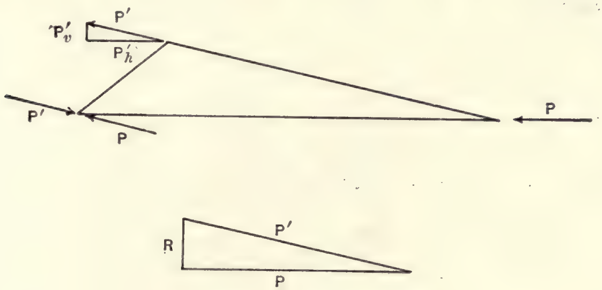


FIG. 315.—Forces on Shaft.

by the spring scale and the area of the piston. The direction and magnitude of the rod pressure are found in a manner similar to that shown in Fig. 315.

The weights of the flywheels are now taken from the determination of chapter VII, and after resolving the connecting

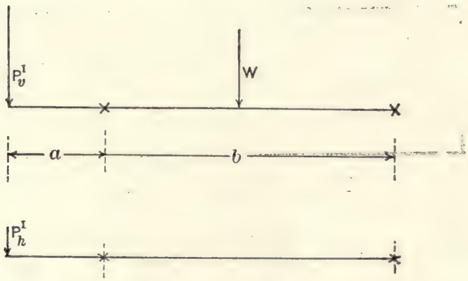


FIG. 316.—Loading Diagram.

rod pressures into vertical and horizontal components, the shaft is treated as a beam, first subject to vertical forces and then to horizontal forces. After the bending moments or bending moment diagrams for these are computed, the resultant moments may be found and the maximum determined. The loading diagrams for a vertical, overhung engine is given in Fig. 316. In these figures the forces are arranged as if they

were acting in the same direction. They may in reality be reversed and when this is the case the sign of the term involving any such should be changed.

The critical points in Fig. 313 are at the left support and under the load.

These may be computed

$$M_v \text{ (at left support)} = P_v'a,$$

$$M_h = P_h'a.$$

At load:

$$M_v = \left(\tfrac{1}{2}W - \frac{P_v'a}{b}\right)\tfrac{1}{2}b,$$

$$M_h = -\left(\frac{P_h'a}{b}\right)\tfrac{1}{2}b.$$

The resultant of M_v and $M_h = \sqrt{M_v^2 + M_h^2}$.

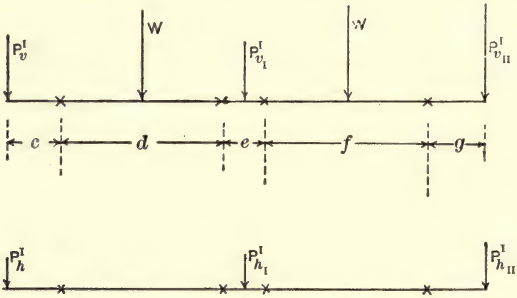


FIG. 317.—Loading Diagram.

The largest of these is the one which fixes the size of the shaft.

In the case of the three-cylinder engine, the shaft is a continuous beam of three spans loaded with concentrated loads and hence the theorem of three moments must be applied to find the moment at the supports.

The general form of the theorem for this case is

$$M'l_1 + 2M''(l_1 + l_2) + M'''l_2 = -P'l_1^2(k_1 - k_1^3) \\ - P''l_2^2(2k_{,,} - 3k_{,,}^2 + k_{,,}^3).$$

The equations for this particular case for which the loading diagram is Fig. 317, are as follows:

$$M_{1v} = -P'_v c,$$

$$M_{1_v} d + 2M_{2_v}(d+e) + M_{3v} e = -Wd^2(\tfrac{1}{2} - \tfrac{1}{8}) - P_{v_1}' e^2[1 - \tfrac{3}{4} + \tfrac{1}{8}],$$

$$M_{2v} e + 2M_{3v}(e+f) + M_{4v} f = -P_{v_1}' e^2(\tfrac{1}{2} - \tfrac{1}{8}) - Wf^2[1 - \tfrac{3}{4} + \tfrac{1}{8}],$$

$$M_{4_v} = +P'_{v_{\prime\prime}} g.$$

From these four equations the values of M over each support may be found and then the shear to the right of each support is given by

$$V_1 = \frac{M_{2_v} - M_{1_v}}{d} + \frac{W}{2},$$

$$V_2 = \frac{M_{3_v} - M_{2_v}}{e} + \frac{P'_v}{2},$$

$$V_3 = \frac{M_{4_v} - M_{3_v}}{f} + \frac{W}{2}.$$

The moments at the load will then be

$$M_{dw} = M_1 + V_1 \frac{d}{2},$$

$$M_{ep} = M_2 + \frac{V_2 e}{2},$$

$$M_{fw} = M_3 + \frac{V_3 f}{2}.$$

This same operation is performed for the horizontal forces, remembering that the terms involving W are zero. From the values of the positive and negative moments the resultants are found and the maximum determined. In this manner determination should be made for various positions of the crank and the maximum of the maxima, ascertained.

This may be the correct method, but a better method would

be to assume the engine fixed on the dead point on one cylinder with full steam pressure, and have the full net steam pressure on one of the others the same while the third has exhaust pressure. With this, work out the equation above. A determination should also be made of full boiler steam on the first crank alone with no pressure on the other pins. In these problems the pressures from the pistons might be considered as acting directly on the pins with connecting rods of $n=6$, as the effect of angularity is not great enough to affect the result.

After making a number of investigations the values of M at the various pins are known.

From the tangential effort diagram, Fig. 283, the twisting moment of any shaft can be found as well as the twist at the various pins. Twisting moment is equal to the tangential effort multiplied by crank radius. The diagram of H.P. cylinder gives the twist from the first crank to the first flywheel, and the third diagram from the L.P. cylinder gives the twist going to the flywheel near it. The amount going to the flywheels is given by the combined diagram, and if one half of this is subtracted from the H.P. diagram the amount which remains shows the quantity going to the second crank from H.P. The twist from the L.P. diagram may be treated in the same manner and the amount received by the second crank from L.P. is given. The quantities from the two ends when added should give the amount from the I.P. diagram. If the amount from the H.P. diagram reaching the second crank is of the same sign as that on the I.P. diagram, the whole twist from H.P. must be transmitted through the second crank pin. If the amount of twist from the I.P. diagram is less than that from the H.P. but of different sign, then the arithmetic difference is transmitted. In this way a diagram may be drawn for the twist which is transmitted across the center crank.

If now from these diagrams the bending moments and twisting moments are obtained at various points in the revolution for the pins and shafts below the wheel, the values of $T_1 = \sqrt{M^2+T^2}$ could be found.

It has been shown by Grashoff that when bending and

torsion occur at the same time in a shaft, the shaft should be designed as subject to a bending moment of M_1 where

$$M_1 = \tfrac{3}{8} M + \tfrac{5}{8}\sqrt{M^2+T^2},$$

when M = bending moment,

T = twisting moment.

M_1 = combined equivalent bending moment

Rankin develops a slightly different formula by considering maximum stress, in place of maximum deformation and arrives at the formula

$$M_1 = \tfrac{1}{2}M + \tfrac{1}{2}\sqrt{M^2+T^2}$$

Mr. J. J. Guest, within the last few years, has examined bodies subject to tension and torsion and finds that the controlling force is not one of tension but of shear, and so he recommends that the combined equivalent shear be used in designing when these two stresses occur. This would give for the moment formula:

$$T_1 = \sqrt{M^2+T^2} = S_s \frac{\pi d^3}{16}.$$

The diameter of the shaft or pin may now be found by this formula.

It is to be noted in passing that when a triple-expansion three-cylinder engine drives a pump or other machine at one end of the shaft, the crank pin nearest the point of driving transmits the sum of the twists from the cylinders placed in front of it.

The investigation for bearing should be made with center cranks at this point, although d' is so large here on account of the bending moment that it will rarely be found necessary to enlarge d' to that given by the formula:

$$d' = \frac{P}{pl'}.$$

Having now the diameter of the crank pin for the various forms of cranks, with the length, the boxes for the ends may be designed.

The proportions recommended by Unwin are given in Fig. 318, where the unit is

$$t=0.08d'+\tfrac{1}{8},$$

when the pin is an overhung pin, but with center pins, the unit is

$$t=0.01\sqrt{P}+\tfrac{1}{4}.$$

The connecting rod proper is subject to stress produced by four causes: 1st, Direct tension from the piston pull when the piston moves toward the head; 2nd, Direct compression on the return; 3d, The bending stresses produced by the column flexure, and 4th, The bending stresses produced by the whipping of the rod as it changes the direction of motion at the top and

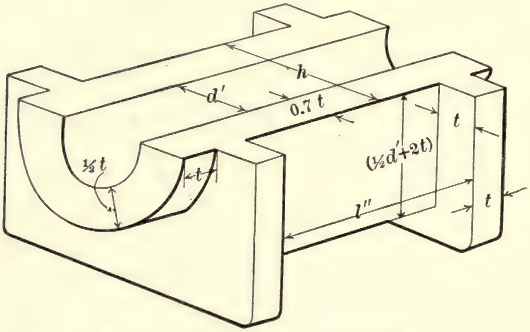


Fig. 318.—Box for Connecting Rod.

bottom positions of the crank in a horizontal engine. The stress in the first case is equal to that in the second, and as the second and third are included in the ordinary column formula, investigation by that formula is sufficient. The fourth stresses are produced by the inertia of the rod, and if m be the mass for any unit length of the rod, and a the maximum acceleration, ma represents the inertia force over that length tending to bend the rod. The acceleration in the rod is a maximum near the crank position 90° from the dead point. The rod is pivoted at the cross-head pin, and hence the acceleration varies as the distance from that point. If the connecting rod is uniform in section, the effect of inertia is the same as if there were a beam loaded with a distributed load which increased from zero at one

end to a certain value at the other, as in the upper part of Fig. 319. The bending moment is a maximum at $0.577l$, and it is at this point the dangerous section for bending occurs.

If the section of the rod is not uniform, but increases toward the crank end, the loading does not vary along a straight line but along some curve, as shown in the lower part of Fig. 319. In this case the dangerous section for the bending is further out.

Now the dangerous section of the column is at $\frac{1}{2}l$, but owing to the bending effect from inertia, consider $0.6l$ as the dangerous section with a rod of varying cross-section for both bending,

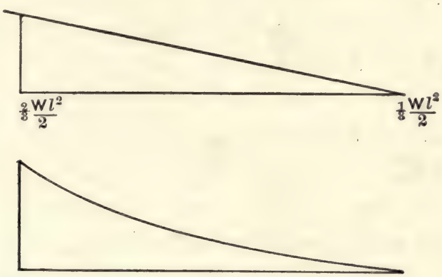


FIG. 319.—Inertia Load on Connecting Rod.

due to the column and to inertia. If the rod is uniform the dangerous section would be between $\frac{1}{2}l$ and $0.577l$.

The stress produced at the dangerous section, treating the rod as a column, is given by the formula:

$$S=\frac{P}{A}\left(1+\frac{Kl^2}{r^2}\right),$$

while the stress by the inertia bending is given by

$$S=\frac{Mc}{I}.$$

Now this latter could only be determined after knowing the shape of the rod, and hence the shape would have to be assumed for a first approximation. In place of this such a value of S in the column formula may be taken that this inertia effect would be covered. If such a value is used and the cross-section determined from formula, the shape is found at the dangerous section, which is considered to be at $0.6l$.

The rod may now be designed. The cross-head end of most rods is made smaller than the remainder of the rod and is designed to take direct tension or compression. This is therefore designed first. If the rod is to be uniform in section this end is not designed, as the size is determined by the cross-section at the center.

The force which this rod may have to stand is, as was mentioned under crank pins,

$$P\frac{n}{\sqrt{n^2-1}}=P'.$$

In many engines the cut off occurs before the crank has reached its mid position, but it is best to use this value as the valve may be reset so that the pressure could be carried to the 90° crank position.

The area at this smallest section is

$$A''=\frac{P'}{4000}.$$

The value $S=4000$ being found, from practice, to be a fair average.

If the rod is rectangular,

$$A''=b''d'',$$

in which b'' or d'' is assumed;

$$\frac{\pi d''^2}{4}=A'',$$

gives the diameter if the neck is circular. The connecting rod, considered as a column in the plane of oscillation, has round ends, while in a perpendicular plane it has fixed ends. In investigating rods of standard engines, considering the dangerous section of rods similar to Fig. 320 as a rectangle, which would circumscribe the section and not as the actual section, it was found that a stress of 4000 pounds used in the column formula would give sections found in modern engines. Hence this includes the effect of whipping as well as the column effect.

Fig. 320.—Standard Rod Cross-section.

Taking now the plane of oscillation, for the section at $0.6l$, the following results:

$$\frac{P'}{A'''}=\frac{4000}{1+\frac{4}{25000}\frac{l^2}{r^2}};$$

or in case the width is the same as the diameter at the smallest section,

$$\frac{P'}{d''d'''}=\frac{4000}{1+\frac{4}{25000}\frac{l^2}{\frac{d'''^2}{12}}}.$$

From these d''' at the dangerous section is found.

Now in the plane perpendicular to the plane of oscillation, consider the rod as a column with fixed ends. This is not strictly true, as there is some play, but since in this plane there is no whipping effect, the value, $S=4000$, will equalize the decrease of strength due to the rod not having strictly fixed ends in that direction.

$$\frac{P'}{d''d'''}=\frac{4000}{1+\frac{1}{25000}\frac{l^2}{\frac{d''^2}{12}}},$$

Solving this for d''', another value is found and the larger of these two is taken.

Barr treats rectangular sections another way. The dangerous section is at the center, and if d''' is made $2d''$, columns of equal strength in the two planes result, hence only solve for d''' once. The formula used is that of Euler, which reduces to

$$d''=c\sqrt{Dl},$$

For steel: $c=0.057$ mean,

$=0.07$ maximum,

$=0.043$ minimum.

In order to overcome the whipping effect d''' is made greater than $2d''$. The mean value is

$$d''' = 2.7d'',$$
$$= 4.d'' \text{ maximum},$$
$$= 2.2d'' \text{ minimum}.$$

The factor of safety here is 13. In this case the taper is obtained by giving the rod at the neck the same cross-section as the piston rod and tapering from the point.

Theoretically the taper need not extend beyond a point at $0.6l$, but in high-speed engines there is some chance for the crank pin to seize and then the connecting rod becomes a cantilever beam loaded at the end which might snap off. Hence in this class the rod taper is usually carried out to the crank end. In slow-speed engines the rod very often tapers each way from the center and is circular in cross-section.

The brasses or boxes *EE*, Fig. 307, are next considered in the light of the above rod design and an endeavor is made to make the distance l'' of Fig. 318 equal to d''. This result would make the forging of the rod simpler, and if it can be done by making the thickness of the flange slightly greater or less by any amount than that shown in Fig. 318, these should be changed to accomplish this end.

Connecting Rod Ends

The connecting rod end is next considered. The sides of the box end carry tension; the end, bending; the rod end, compression. The bolts of the marine end carry tension; the keeper, bending, and the club end, compression. The strap end has to be designed for tension and bending in the strap, compression in the rod end, and shear and compression in the gib and key.

Box end. (Figs. 307—308.) Assume the diameter of the bolt attached to the wedge $\frac{1}{2}''$ or $\frac{3}{4}'' = d^{iv}$.

Find b from brass and if possible make it equal to the thickness of the rod d''. Then

$$2t'(b-d^{iv})S_t=P',$$

$$t'=\frac{P'}{2(b-d^{iv})S_t}.$$

The rod end could be designed as a beam from the formula

$$\tfrac{1}{8}P'\times h=S_t\frac{b(t^{v})^2}{6}.$$

Since the bending of this rod end would send the load to the corners it may be well to neglect the design of this as a beam, determining only the thickness at the corners for shear and making $t^{v}=1\tfrac{1}{2}t^{vi}$ and using a circular arc for the ends, as shown, for appearance,

$$t^{vi}=\frac{P'}{2bS_s}.$$

The wedge should be sufficiently high for crushing and also about three-fourths the height of the opening of the rod in length so that the brass will be well supported.

$$h'=\frac{P'}{bS_c}>\tfrac{3}{4}h=\tfrac{3}{4}(d'+1.4t).$$

The bolt at the end of rod shown in Fig. 308 supports all of the load, hence

$$\sqrt{\frac{P'}{2S_s\frac{\pi}{4}}}=d^{v}.$$

This should be designed for crushing and the fork end should carry tension

$$t^{vii}=>\frac{P'}{2S_cd^{v}},$$

$$t^{vii}=>\frac{P'}{2S_t(b-d^{v})}.$$

This may require a thickening because of the large bolt used. Such a result is shown in Fig. 308.

Marine end. Bolt area at root of thread, $=\dfrac{P'}{2S_t}=\dfrac{\pi d_1^2}{4}$.

Breadth keeper, $b_1=d_1=$amount from box design.

$$l_1=d'+2t+d_1(t \text{ from brass}).$$

Thickness keeper, $\frac{1}{8}P'l_1=\dfrac{S_tb_1t_1^2}{6}$.

Length of keeper, $l_{,}+2d_1=l_{,,}$.

Crushing, $P'<S_cl_{,,}b_1$.

Strap end. Key and gib design:

$$2S_sb'''l'''=2S_ct'b'''=2S_t(b-b''')t'=P'.$$

$$b'''=\frac{S_t}{S_c}(b-b''').$$

$$\frac{S_t}{S_c}=\frac{1}{2} \text{ (in general)}$$

$$b'''=\tfrac{1}{2}(b-b'''),$$

$$\tfrac{3}{2}b'''=\tfrac{1}{2}b.$$

$$b'''=\tfrac{1}{3}b.$$

$$t'=\frac{P'}{2S_cb'''}.$$

$$l'''=\frac{P'}{2S_sb'''}.$$

$$t^{\text{viii}}=t'$$

$$l^{\text{iv}}=0.7t'.$$

Taper of key $\frac{3}{4}''$ per foot.

The key and gib are each made $\frac{1}{2}l'''$ in width at top, when key is first entered.

In the rod end the height t is usually much greater than $2t$, and hence there is no need of investigating for crushing between the key and the rod end. If this were not so an investigation would be made.

The end of the strap is designed in the same manner as the end of the box.

BEARINGS

The bearings of the shaft are designed for sufficient bearing area to care for the load. Using the vertical shears at each side of the bearing in the continuous beam design the algebraic difference of these would be the reaction, and if this is found in the horizontal and vertical planes the resultant R is given by

$$R=\sqrt{R_h^2+R_v^2}.$$

As an example in Fig. 317, the shears at the left of the supports have been found as V_1, V_2, V_3, etc. On the right of these supports the shears will be

P_v' at right of first.
V_1 at left of first.
V_1+W at right of second.
V_2 at left of second
$V_2+P^{v\prime\prime}$ at right of second, etc.

The first reaction from the vertical forces will be $P'-V_1$. The various reactions from the vertical forces will be found in the same manner; in like manner the horizontal reactions and from them the resultant.

In Fig. 316 the reactions are:

Right horizontal, $-P_h'\dfrac{a}{b}$.

Right vertical, $-P_v'\dfrac{a}{b}+\frac{1}{2}W$.

Left horizontal, $P_h'\dfrac{a+b}{b}$.

Left vertical, $P_v'\dfrac{a+b}{b}+\frac{1}{2}W$.

Right resultant, $=R=\sqrt{P_h'^2\dfrac{a^2}{b^2}+\left(-P_v'\dfrac{a}{b}+\frac{1}{2}W\right)^2}$.

Left resultant, $=R=\sqrt{\left(P_h'\dfrac{a+b}{b}\right)^2+\left(P_v'\dfrac{a+b}{b}+\frac{1}{2}W\right)^2}$.

Having the pressure allowable, the diameter of the shaft at the journal (from stress design) and the load on the bearing, the length is given by

$$l'' = \frac{R}{pd''}.$$

For p Unwin recommends from 150 to 450, the larger

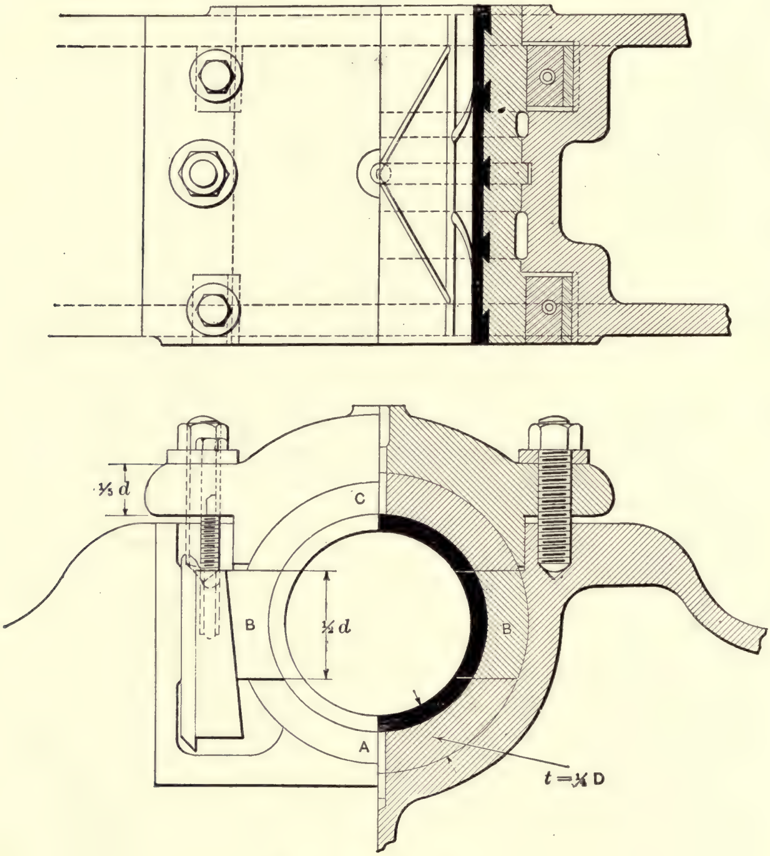


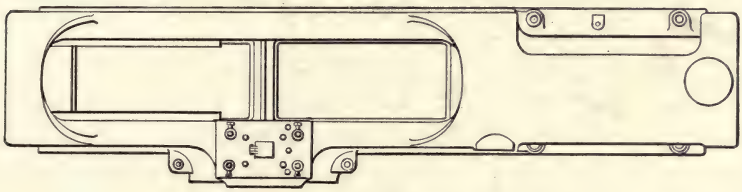
Fig. 321.—Main Bearing.

values being used with slow-running engines, in which there may be a reversal of load as near the stroke end. For fast engines with shafts carrying a steady load in one direction the low

value should be used. 200 pounds is an average value which may be used.

The bearings of large pumps are shown in Figs. 321 and 322. They are usually made in four parts:—one bottom brass *A*, two cheek pieces *B*, and one top piece *C*. This is done so that wear may be taken up horizontally and vertically.

In building the boxes for the main bearing it is well to make the lower section circular at the bottom, so that when necessary to renew this section it may be taken out by blocking



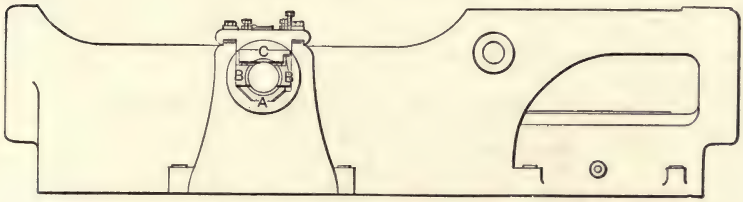


FIG. 322.—Bearing and Frame.

up the shaft and twisting the box around. Of course such a section does not resist the effect of friction to turn the brass, nor can a shim or liner be introduced beneath the circular bottom as it could be with the square base. If the box can be moved along the axis of the shaft by the removal of bolts or caps this would enable one to remove the box easily, and in this case a flat base could be used, as in Fig. 322.

The proportional parts of these boxes and bearings are shown in Figs. 321 and 322. The boxes are to be grooved so that oil will be distributed and in many cases it will pay to deliver oil under pressure to the main bearings. In any case

a steady oil stream should be used, the oil being filtered after using.

The thickness and projection of flanges on bearing brasses are made $\frac{1}{8}l$.

If desired the surface of the central portion of the box which supports the box may be cut away throwing the load to the ends of the box. When this is done the width of the removed portion is $\frac{1}{3}l$.

Where collars or crank bosses wear against the flange of the box a slight amount of play is given to the shaft by leaving a small amount of clearance.

The bolts holding the top cap should be strong enough to hold the cap in place, and also in the case of horizontal engines, to carry part of the thrust of the piston across the top of the opening in the frame. This may be done by projections on the side of the top cap, as shown in Fig. 322, when the bolts are only used to keep the cap in place. The exact amount of this force required to hold the cap in place cannot be computed, so that this part of the design will be empirical, and the following number of bolts will be used.

Shaft Diameter.	No. of Cap Bolts.	Diameter.
6″	2	$1\frac{1}{4}$″
8	2	$1\frac{1}{2}$
12	4	$1\frac{1}{2}$
18	4	$1\frac{3}{4}$
24	4	2

These bolts are often made T-headed and fitted into pockets in the sides of the spaces left to receive the boxes.

The wedge bolts for lifting are usually two in number with one or two set screws in addition, so that the bolts may be used to raise the wedge while the set screws are used to force it down after the bolts are loosened, or they may be used to hold the cheek piece in place by tightening them up against the pressure from the bolts. These bolts and set screws may be made $\frac{3}{4}$″ in diameter for boxes of shafts up to 12″ and 1″ beyond that.

Fly Wheel

The fly wheel is usually made as shown in Figs. 323 and 324. That shown in Fig. 323 is in two sections while that of Fig. 324 is in five sections.

The two-section wheel is bolted together at the shaft, and at the rim it is held by a link *A* which passes around projections *BB* on each end of the sections. There are two, three or four of these rings used at each joint. These projections are usually

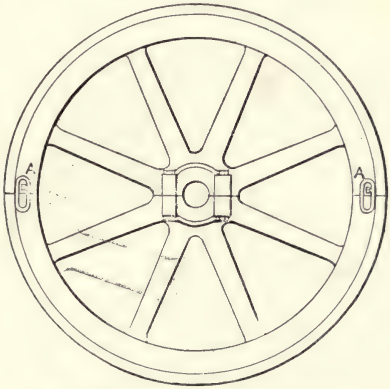


Fig. 323.—Two-part Fly Wheel.

made by forming a ring shaped cavity in the face of the rim so that the ring is flush with the surface.

In Fig. 324 the wheel consists of five sections bolted at the center to two flanges *AA*, called spiders or centers. The sections are held together at the rim by T-headed links *B* which are placed in cavities in the rim. These links and those of the previous figure are usually placed into position by heating, the length, when cooled, being enough less than the length between ends of projection that the extension to the proper length would produce the force desired to hold these together under the centrifugal load.

The fly wheel is used at times as a driving wheel for gears and belts, but in any case it is subject to stresses of a complex nature. These will be examined in parts.

Consider the rim of a wheel as solid and free from spokes; on each square inch of area of this rim there is a centrifugal normal force of

$$F = \frac{tw}{g} \frac{V^2}{r},$$

where t = thickness in inches,
w = weight of one cu.in.,
V = velocity of center of this rim,
r = radius of center of rim $[\frac{1}{2}(r_{\text{out}} + r_{\text{in}})]$ in ft.

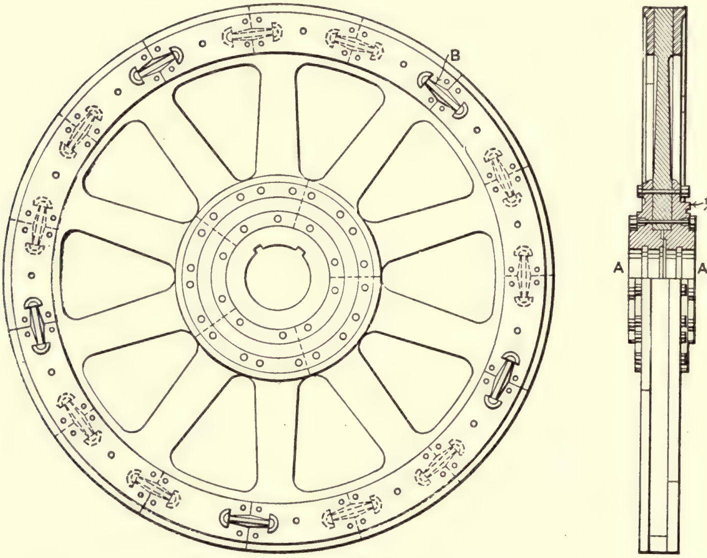


FIG. 324.—Sectional Fly Wheel.

Now the rim under this load may be considered as a thin cylinder subject to a hydrostatic pressure, and hence the formula: $pd = 2tS$ holds.

In this case

$$p = F,$$
$$d = 2r \times 12,$$
$$t = t,$$
$$S = \text{unit stress}.$$

Hence

$$\frac{tw}{g}\frac{V^2}{r}\times 24r = 2tS$$

or for the total breadth

$$\frac{tw}{g}V^2\,12b = tbS = T$$

When T is the total pull causing rupture.
The unit stress S becomes

$$S = \frac{12wV^2}{g}.$$

That is, the unit stress in a rim under centrifugal force when rotated at the velocity V is independent of the radius of the wheel and size of the rim, and is equal to

$$\frac{12wV^2}{g} = \tfrac{3}{8}wV^2.$$

For cast iron $w = 0.26$ lb. per cu.in.

$S = \frac{1}{10}V^2$ approximately.

This stress of $\frac{3}{8}wV^2$ produces an elongation in the radius of $\frac{3}{8}\frac{wV^2r}{E}$, and if spokes are used the rim will take the exaggerated form shown in Fig. 325. The spokes will be stressed by the rim pull and in turn this will cause a distortion, as shown, so that at points AA there will be no bending, but at the other portions of the rim bending forces will result. These bending forces have maximum values at b and c, the positive maximum being at c, while at b the negative maximum occurs.

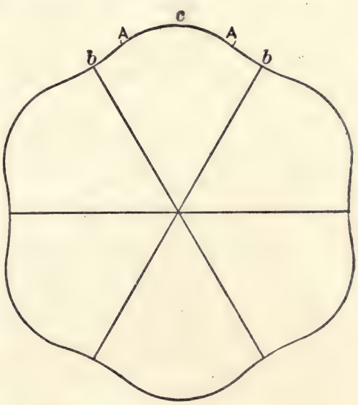


FIG. 325.—Rim Shape.

The variation of these stresses has been investigated theo-

retically by Unwin in his "Machine Design," part II, pp., 293–301, and he derives the following formulæ:

$$F=\tfrac{2}{3}\frac{G}{g}\frac{V^2}{\frac{1}{A_1}+\frac{1}{2A\alpha}}=\tfrac{2}{3}\frac{G}{g}V^2\frac{2A_1A\alpha}{A_1+2A\alpha},$$

$$M=\frac{FR}{2\alpha}-\frac{FR}{2}\,\frac{\cos(\alpha-\phi)}{\sin\alpha},$$

$$T=\frac{G}{g}AV^2-\frac{F\cos(\alpha-\phi)}{2\sin\alpha},$$

$$S=-\frac{F}{2}\frac{\sin(\alpha-\phi)}{\sin\alpha}.$$

Where F = pull on one arm in pounds.
M = bending moment at any section of the rim in pound feet.
T = tension at any section of the rim in pounds.
S = shear at any section of the rim in lbs.
$\alpha=\frac{1}{2}$ the angle between the arms in radians.
G = weight of metal per cu.ft. in lbs.
$g=32.16$.
A = area of cross-section of rim in sq.ft.
A_1 = area of cross-section of spoke in sq.ft. (mean).
V = velocity of center of rim in ft. per sec.
R = radius to center of rim in ft.
ϕ = angle in radians to any point measured from center line of spoke.

The experimental investigation of the resistance of fly wheels to breaking or exploding, as it has been called, has been made by Dean Chas. H. Benjamin of Purdue. In these investigations reported to the American Society of Mechanical Engineers, (Vols. 20 and 22) wheels of different designs, of reduced size, were run until rupture occurred. At this point the speed was noted, and from it the coefficient $\frac{V^2}{10}$, derived above, was computed.

In the case of a 15-inch solid wheel, the speed at rupture was 390 feet per second (6000 R.P.M.) and the value of $\frac{V^2}{10}=14{,}500$. The strength of the cast iron from which this wheel was made was 19,000 lbs. per square inch in direct tension and 39,000 was the modulus of rupture. With a flanged jointed wheel of 24 inches diameter, the speed at rupture was 190 feet per second, with $\frac{V^2}{10}=3600$ lbs. per square inch, and S_t was 19,600. With a linked joint similar to Fig. 323 the speed at rupture was 305 feet per second and the value of $\frac{V^2}{10}$ was 9300.

From these and other tests it is seen that the solid rim wheel will develop slightly less than its tensile strength, while the linked wheel will have about two-thirds the strength of the solid wheel, and the flanged jointed fly wheel but one-quarter the strength.

It may be assumed then that the rim is designed to take the centrifugal tension.

There are other stresses brought on the rim and arms when the speed changes. Consider a wheel in which the arms and rim sections are not united. When the engine is suddenly speeded up the inertia of these sections acts as a load on the sections and bends the arms, as shown in Fig. 326, where the rim sections are joined. This same action causes the rim to take the form shown. This action complicates the formulæ given by Unwin, and to cover them low stress should be used, and then the rim is designed for the centrifugal load, while the arms are designed to take the bending due to driving or to inertia.

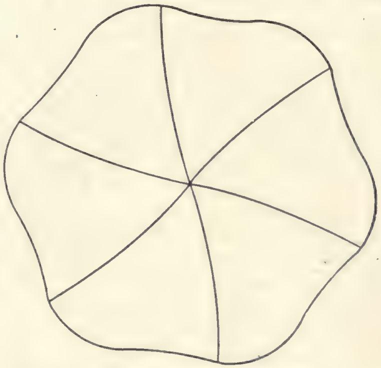


FIG. 326.—Effect of Bending.

The twisting moment or tangential effort diagram, Fig. 283,

gives the moment for which the arm must be designed. Let h be the greatest height of the combined tangential effort diagram which shows the amount of effort given to the two fly wheels. The moment exerted by this is

$$\tfrac{1}{2}h\times \text{ scale}\times\text{L.P. piston area }\times\text{crank arm}=M.$$

This is the maximum moment during one revolution after the engine has been started, but if steam is admitted suddenly in starting the pump, the entire energy from the piston will be transmitted to the fly wheels, and it is well to design the arms for that contingency. This represents the maximum which could ever come on the arms. This gives

$$M_{\text{max}}=\tfrac{1}{2}\frac{\pi d^2}{4}\,d_{\text{h.p.}}\,p_{\text{boiler}}\times\text{crank arm}.$$

This moment is divided between m arms, hence

$$\frac{M_{\text{max}}}{m}=S_t\frac{\pi bd^2}{32},$$

for the elliptical arms of diameters b and d.

This is really the moment and size at the center of the shaft and the arms are tapered on each diameter to the rim to two-thirds these dimensions.

The rim of a solid wheel is sufficiently strong, provided V be determined by the formula

$$\frac{V^2}{10}<\text{ safe stress.}$$

There is no way of making this stronger if the material is fixed. For high speeds steel may be used to increase the right hand side and so permit a higher velocity.

The links used to join the sections are subject to a load

$$\tfrac{3}{8}wV^2\times\text{area of section.}$$

Hence, if there are n sections of link, each of area a at the section,

$$\tfrac{3}{8}wV^2A=naS_t.$$

From this a can be found, and from it the cross-section dimensions of the link.

The bolts at the center of the wheel, as in Fig. 323 or 324, should be large enough, if possible, to carry the section if the links should break. To find the radial force produced by a section of the wheel, it is necessary to find the center of gravity of the section since

$$\text{C.F.} = \frac{W}{g}\frac{V^2}{R},$$

where W is the weight of the rotating body, V the velocity of the center of gravity and R the radius to that point.

The center of gravity of a section of a ring of angular extent α and of thickness t and mean diameter D, is

$$R = \frac{\sin\frac{1}{2}\alpha}{\alpha}\left[1 + \frac{1}{3}\left(\frac{t}{D}\right)^2\right]D.$$

This then gives the centrifugal force from the rim of a section, to which may be added that of the arm by considering the center of gravity to be at the center of the length of the arm. Calling these forces $F_{\text{c.f.}}$, the following gives the size of the n' bolts in tension or the n'' bolts in double shear to resist this,

$$\frac{F_{\text{c.f.}}}{2n''S_s} = a = \frac{F_{\text{c.f.}}}{n'S_t}.$$

The center discs for the built up wheel and the hub of the split wheel are usually of empirical design, and in the figure the unit has been taken equal to $\sqrt{\text{rim area.}}$

The fly wheels should be keyed to the shaft. The force coming on the key is, $\dfrac{M_{\max}}{\frac{1}{2}\text{ shaft diam.}}$.

$$\frac{M_{\max}}{\frac{1}{2}\text{ shaft diam.}} = F_k = S_s lb = S_c \tfrac{1}{2} dl,$$

l = length of key in inches,

b = breadth of key in inches,

d = depth of key in inches.

The balancing of fly wheels of pumps is a simple matter, for the speeds are low, and all that is necessary is to have the wheel in static balance. The balancing of the reciprocating parts is not important in the slow-running pumps. For a detailed discussion of balancing of the reciprocating and rotating parts the reader is referred to "The Balancing of Engines," by W. E. Dalby.

Frames

The frames of pumps are of different forms. Fig. 327 shows a type in which the *A* frames supporting the cylinders are

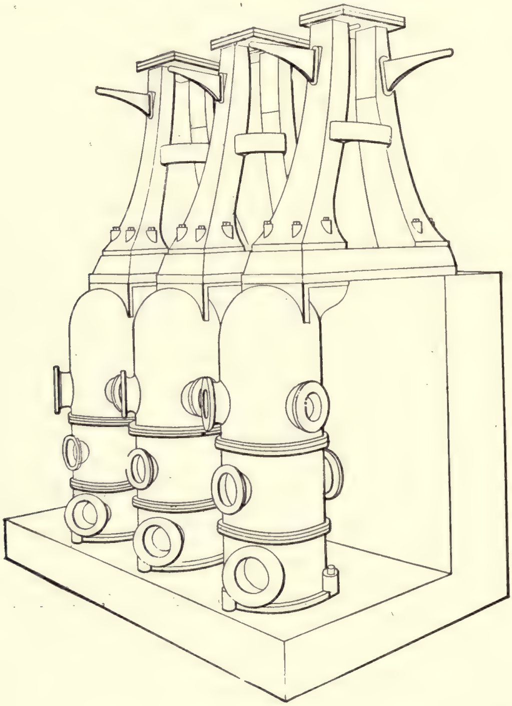


Fig. 327.—Frame for Vertical Engine.

carried on bed plates. The bed plate is carried on the air chambers on one side and on masonry on the other. The design of such structures is from experience. The thickness of the metal usually being about one inch and the sizes of the columns are

fixed by what has been done. The load coming on the frame, if inertia and friction are neglected, is equal to the net steam pressure multiplied by the area of the piston. This force has to be transmitted down in case of the upstroke, and on the downstroke the pressure is upward and is resisted by the weight of this section of the engine and the weight of the masonry when the pump is carried below on the foundation. In the case of a single-acting pump the major part of the force on the upstroke is transmitted to the main bearing and this causes a bend in the frame of the overhung crank engine. This bending moment is equal to

$$P \times (\text{distance } a, \text{ Fig. } 316).$$

The moment is resisted by the frame. On the downstroke the pressure from the steam is balanced by the water pressure, which is central, and so there is no tendency to bend. To cut down vibrations in the frame cross pieces or knees are put in at frequent intervals.

Fig. 346 shows a type of frame in which cast iron columns are used to carry the weight to the lower foundations, while in Fig. 349 the weight is carried by the air chambers entirely, a method which gives good results, as it clears space around the base, facilitating repair, examination or operation. By examining the various figures of the horizontal and vertical pumps, the methods of general practice may be seen.

The operating floors are carried at different levels from beams bolted to the frames. These should be designed to carry about two hundred pounds per square foot.

The guide surfaces are carried in the frame, and if not a portion of the main casting the connections should be made to carry the load

$$\frac{P'}{n}$$

The foundations are designed from experience. There are two principles to consider: First, the foundation should be large enough to properly distribute and carry the load. and second,

there should be sufficient mass to reduce vibrations of the machine.

The area of the base of the foundation should be such that the weight carried from the engine and foundation is 3000 to 6000 pounds per square foot. The first value is for good clay soil, the latter for a compact gravel. To get this it may be necessary to spread the foundation, and to do this a batter of about one foot in two or three feet of height will be found proper to develop the strength of the projecting part. The foundations may be made of concrete of proportions: one part cement, two parts sand and five parts broken stone; or hard burned bricks, set in cement mortar, may be used. These brick should be thoroughly wet when laid and the joints should be grouted with a watery mixture of cement mortar.

The amount of material to be placed in the foundation to reduce vibrations is a matter of experience. In a slow running machine there is practically no need of heavy foundations, and when the engine is even of high speed and properly balanced, there is no need of making the foundation very thick. A thickness of three or four feet is sufficient for horizontal machines, and for vertical machines the thickness will be determined by the amount of height necessary to give the proper spread to the base.

When the engine is to be placed over a very soft soil, it is necessary to drive piles for the support of the masonry. These are laid out so as to support a given load, say forty tons apiece, and when driving, the pile is driven until the amount of penetration under a given blow of the hammer shows that it will carry the desired load. If this load can not be obtained the piles must be placed close together. At times pier holes may be sunk over the engine base area to good soil or rock and the main foundation can be carried on the piers built in them.

A formula recommended by Baker for bearing power of a pile is

$$P=100\left(\sqrt{Wh+(50d)^2}-50d\right),$$

where P = supporting pressure in tons,
W = weight of ram in tons,
h = height of fall in feet,
d = penetration of pile in feet at last blow.

The piles should be of good quality, of sound white oak, not less than 10 inches in diameter at the smaller end and 14 inches at the larger. They should be straight grained and have all bark removed. After driving they are cut off below the permanent water line, and then concrete is put around the end, making a solid bed.

Special Steam Piping and Valves.

The steam piping used on the engine should be designed so that the velocity of the steam is 6000 feet per minute. This is a simple method and gives good results. There is a method in which the pipe is designed to give a certain discharge when the drop in pressure is assumed. The formula developed by Prof. Carpenter for this (A. S. M. E., vol. XX, p. 342) is

$$p=\frac{1}{20.663}k\left(1+\frac{3.6}{d'}\right)\frac{w'^2L}{Dd'^5},$$

where p = loss of pressure in lbs. per sq.in.,
k = a constant = 0.0027,
d' = diameter of pipe in inches,
w' = flow of steam per minute in pounds,
L = length of pipe in feet,
D = weight of one cu.ft. of steam.

This may be used when long pipes are employed, but the rule above for the area will give satisfactory results in pumping stations, and even 8000 feet per minute may be used when necessary for large pipes.

The steam lines should be made of heavy full weight pipe and tested to stand 250 pounds per square inch. In this work most of the pipes are large and the joints are made by flanges.

Fig. 328 shows the proportion of the standard screwed flange, while Fig. 329 shows a similar flange with a caulking recess on

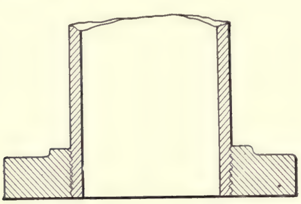


FIG. 328.—Flange.

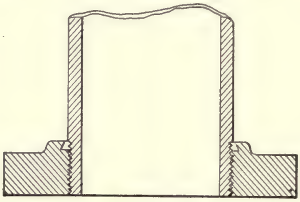


FIG. 329.—Screwed Flange with Caulking Recess.

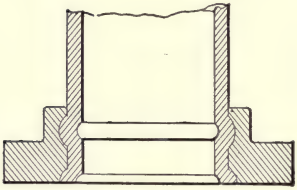


FIG. 330.—Shrunk Flange.

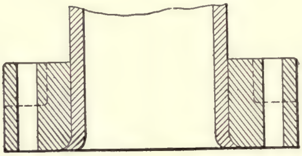


FIG. 331.—Rolled Joint.

the back which may be filled with a metal for caulking. This is not always advisable. These two figures show the pipe screwed on the flange. This is done so that the pipe projects a slight distance beyond the flange when it is tightened to its full extent, and then the projection is turned off flush with the flange. The thread of the pipe is cut deep to accomplish this result.

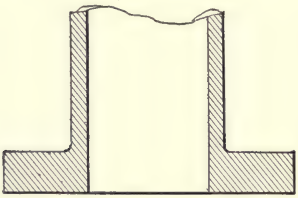


FIG. 332.—Welded Flange.

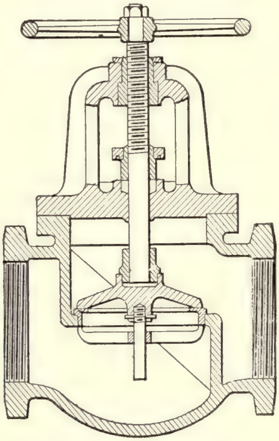
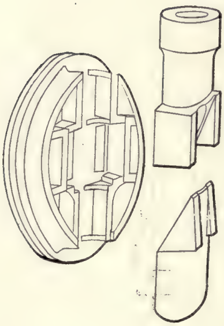


FIG. 333.—Globe Valve.

Flanges are now shrunk on the pipe, Fig. 330, after which the end of the pipe is peened into a cavity left in the flange, and

then the end is turned flush with the flange. The Crane Company sometimes roll the pipe into grooves turned in the flange, as shown in Fig. 331. This operation is similar to the expanding of boiler tubes into the heads of boilers.

The welded flange, Fig. 332, is a new form of flange connec-



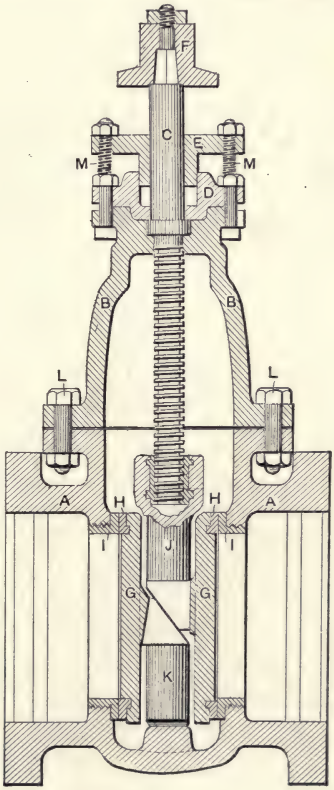


Fig. 334.—Ludlow Gate Valve.

tion for joining pipes, and according to the manufacturer it is meeting with favor for high pressure work.

The dimension of these flanges, the number and sizes of the bolts, the size of various fittings, bends and other specials for pipe work, are to be found in the catalogues and on the dimension sheets of Crane, Walworth, or the other manufacturers of pipe fittings.

The valves used are of the globe or gate types. The globe valve shown in Fig. 333 is of the general form with an outside yoke, and the type of gate valve shown in Fig. 334 is often employed. The dimensions of these valves are found in tables furnished by the manufacturers, so that in laying out work the engineer may know how much to allow for these fittings.

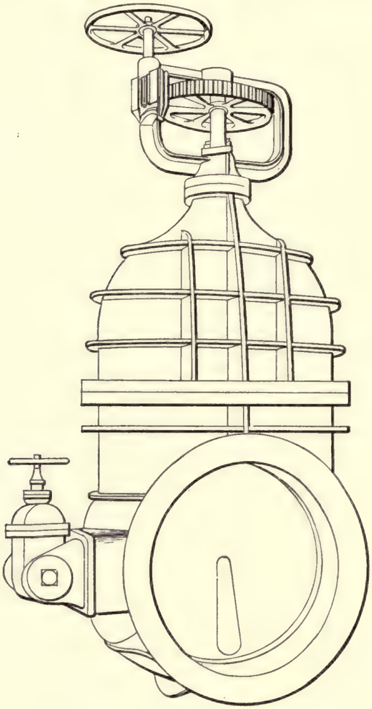


FIG. 335.—Crane Gate Valve.

The valves may have screw ends, Fig. 333, or flange ends, Fig. 334, while for water pipe bell ends are used. The large gate valves, Fig. 335, used on water lines are built with gearing to turn the spindle.

CONDENSERS.

The condensers used are either jet or surface condensers. When the jet condensers are used, the volume of the condenser head should be one third of the volume of the low

pressure cylinder. The pipe through which the water enters the head should be such that the velocity is $2\frac{1}{2}$ feet per second. That is,

$$A=\frac{W_{c.w.}144}{62.5\times 2\frac{1}{2}}.$$

A = net area pipe in sq.in.

$W_{c.w.}$ = water required per sec. in lbs.

The surface of the surface condenser should be determined so that the condensation may be accomplished by the transmission of 500 B.T.U. per hour per sq. ft. of surface per degree difference in temperature.

This gives

$$S=\frac{W(H-q_0')}{500\left(t_s-\frac{t_0+t_i}{2}\right)},$$

where

S = surface in square feet,
W = weight of steam condensed per hour,
H = heat content of exhaust steam
$=q+xr$ or approximately $q+r$,
t_s = temperature of steam in condenser,
t_0 = temperature of condensing water leaving condenser,
t_i = temperature of condensing water entering condenser.

In the *Journal of the American Society of Mechanical Engineers* for November, 1910, Mr. Geo. A. Orrok gives a résumé of a greater number of experiments and theoretical papers on the transmission of heat by condenser tubes as well as the results of a great number of experiments of his own and the deductions drawn from all of these. In this paper Orrok derives the equation,

$$S=\frac{8W\cdot G}{K}\left[(t_s-t_i)^{\frac{1}{8}}-(t_s-t_o)^{\frac{1}{8}}\right],$$

$$K=630c\rho^5\mu\sqrt{V_w}.$$

S = square feet of condenser surface.
W = steam condensed per hour in pounds.

G = condensing water per pound of steam in pounds.

t_s = temperature of steam.

t_i = temperature of condensing water at inlet.

t_o = temperature of condensing water at outlet.

c = cleanness factor, varying from 1.00 with clean tubes to 0.5 with dirty tubes.

μ = material coefficient.

= 1.00 for copper.

= 0.98 for Admiralty tubes.

= 0.97 for Admiralty aluminium lined.

= 0.92 for Admiralty oxidized (black).

= 0.87 for aluminium bronze.

= 0.80 for cupro-nickel.

= 0.79 for tin.

= 0.75 for zinc.

= 0.74 for Monel metal.

= 0.63 for Shelby steel.

= 0.55 for Admiralty badly corroded.

= 0.47 for Admiralty vulcanized inside.

= 0.25 for glass.

= 0.17 for Admiralty vulcanized both sides.

ρ = ratio of the steam pressure corresponding to the temperature to the total vacuum pressure $= \dfrac{P_s}{P_c}$.

V_w = velocity of water in tubes in feet per second.

Now θ = mean temperature difference in condenser

$$= \left[\frac{\frac{1}{8}(t_0 - t_i)}{(t_s - t_i)^{\frac{1}{8}} - (t_s - t_0)^{\frac{1}{8}}}\right]^{\frac{8}{7}}.$$

In practice, when $t_i = 70°$ F., $t_0 = 90°$ F., $t_s = 98°$ F., $\theta = 16.5$. If there is a 28-inch vacuum, and the tubes are medium clean Admiralty tubes in which the water is flowing at 4 feet per second, the following value is found for S for each 1000 pounds of cooling water per hour:

$$K = 630 \times 0.85 \times \left(\frac{.887}{.943}\right)^5 \times 0.98 \times \sqrt{4} = 773,$$

$$S = \frac{1000 \times 8}{773}\left[(98-70)^{\frac{1}{8}} - (98-90)^{\frac{1}{8}}\right] = 2.3.$$

Sometimes the surface is determined by the I.H.P. of the engine. The following table is taken from Peabody's "Thermodynamics of the Steam Engine":

Absolute Terminal Pressure in lbs. per sq. in.	Sq. ft. per 1 H.P.
20	1.71
15	1.57
12½	1.50
10	1.43
8	1.37
6	1.30

The air pump of the jet condenser is made sufficiently large to care for the condensed steam, the condensing water and the air. For this purpose it is made so that its displacement is about forty times the volume of the condensed steam. For wet air pumps used with surface condensers, the pump handles only the condensed steam and air, and therefore the volume displaced is reduced to one-half of the former amount or twenty times the volume of the condensed steam. A better method of determining the volume will be given in Chapter XI.

The quantity of cooling water to be supplied these condensers and to be cared for by the pumps is given by the equation:

$$W_{\text{c.w.}} = \frac{W(H - q_0')}{q_0 - q_i},$$

$W_{\text{c.w.}}$ = amount of cooling water to be used with W lbs. of steam;

H = heat content of exhaust steam $= q + xr$ assumed to be $q + r$;

q_0' = heat of liquid of condensed steam;

q_0 = heat of liquid of condensing water leaving;

q_i = heat of liquid of condensing water entering.

CHAPTER IX

TEST OF PUMPING ENGINES

Duty Trial

Engines are usually tested for steam consumption; that is, the number of pounds of steam per horse-power-hour, but with pumps, their duty is usually determined. Duty is the number of foot pounds of useful work done by one thousand pounds of dry steam or one million British thermal units used by the pump. The problem, then, is to find the number of foot pounds of work done by the pump, and the number of heat units used by the pump. Duty meant originally the amount of work per hundred pounds of coal, so that different pumps could be compared. As coals varied this did not have a definite meaning, and so a more definite unit was desired. The value 1,000,000 was taken because on an average one pound of coal gives 10,000 B.T.U. to the water in the boiler, and hence 1,000,000 B.T.U. is equivalent to an average of 100 pounds of coal, while 1,000 pounds of steam can be assumed to be evaporated by this coal, as each pound of water requires about 1,000 B.T.U. to evaporate it.

In considering the useful work done, the work necessary to bring the water to the pump and that necessary to send it from the pump to the reservoir must be included. The work necessary to overcome the friction in the pump itself, i.e., through valves and losses of sudden contraction, etc., is not included, for these are due to the pump. With this in mind it is seen that the useful work is

$$W = wQ\left(144\frac{p_1}{w} + h + \frac{p_2}{w}144\right),$$

where W is total useful work; w the weight of one cubic foot of water; Q the total number of cubic feet pumped; p_1 the pres-

sure head in the force main in pounds per square inch; p_2 the vacuum in the suction pipe in pounds per square inch, and h the distance in feet between the force gauge and the suction gauge; or

$$W = Q(144p_1 + wh + 144p_2)$$

where the symbols mean the same as before.

The value of w can be found from the temperature of water pumped, and in important cases a known volume of water should be weighed to find the effect of suspended matter.

The quantity of water is usually determined in one of five ways: first, by displacement; second, by weirs; third, by meters; fourth, by weighing; fifth, by measuring.

First, the diameter of the plunger is measured and also the length of the stroke. These, with the number of the strokes, give the displacement, or the quantity of water pumped if there is no leakage, or slip, as it is sometimes called. This slip occurs around the piston or plunger in an inside packed piston; but with the method of outside packing used to-day this leakage is nothing and the only other place for leakage to occur is past the valves. In order to determine the leakage around the piston when inside packed, and the valves when seated, standard methods are used. (See Carpenter, "Experimental Engineering," page 560, and Kent, "Pocket Book," page 611.) The leakage past the valves in seating cannot be ascertained, except by determining Q by one of the other methods.

In some pumps the stroke of the plunger is not definite, and hence the stroke should be measured continuously. If conditions are nearly constant the stroke will not vary much, and hence readings of length taken at regular intervals are made. This is not absolutely correct, and hence some form of registering apparatus should be employed. (See A.S.M.E., vol. 12, page 981.)

Having now the areas, stroke and leakage, the amount pumped may be computed

$$Q = nNLA \cdot \left(\frac{100 - \% \text{ slip}}{100}\right).$$

Second, for this method the constants for the weir and also the hook gauge must be known. Attempt should be made to reduce the fluctuation of the water surface, and the gauge should be read in a box on one side of the flume.

Third, since the introduction of the Venturi meter this instrument can be used to advantage in pump tests, as its error is small. The Pitot tube also serves as a most accurate method of determining the quantity if a traverse is made by a tube of small size pointing in direction of the current when the static pressure is taken from holes in the pipe walls.

Fourth, for small pumps the water pumped can be actually weighed, thus giving the exact amount pumped.

Fifth, cisterns and reservoirs may be used when their contents are known, but here allowance must be made for the evaporation from the surface and the effect of wind.

The quantity p_1 is determined by means of a pressure gauge which must be tested. (Be sure to take maker's number of all instruments used and position where used.) Since the pressure in the force main varies considerably, the gauge cock is throttled off, or an air chamber is introduced between the gauge and the force main which will reduce the fluctuations. p_2 is usually determined by a mercury tube, and hence must be reduced from inches of mercury to pounds per square inch. This pressure is greater than the head between the points where measured and the water in the forebay, as it is equal to the velocity head and the head of friction loss, in addition to the lift. This should be included, as it is not on account of the pump that such losses occur, but on account of the location of the pump demanding a certain length of pipe.

The difference of level between the suction gauge and force gauge must be taken, because the water is raised through this distance, but the distance is not given by either of the pressure readings.

Having these, the useful work can be found. If indicator cards are taken from the cylinder, they give the work done on the water, and the difference between this and the useful work

just determined, would give the losses due to leakage and the friction of the water in the pump.

The readings of pressure and the revolution counter are taken at regular intervals, and the average result used.

To determine the quantity of heat used by the engine, the quantity of heat in each pound of steam used in the engine must be found as well as the total number of pounds of steam used. The best way to determine the quantity of steam used is to condense the steam in a surface condenser and weigh it. To this must be added the amount consumed in the steam jackets and the superheaters.

The water coming from the jackets and superheaters is at such a temperature that it would partially evaporate if allowed to discharge into the air before being cooled. There are two methods of weighing this: one consists of allowing the hot water to discharge into a tank partially filled with cold water; the other is to conduct it through a coil surrounded by cold water and discharge into tanks.

As condensers attached to pumping engines are often jet condensers the above method is impossible, and so the following method must be used: A number of boilers sufficient to run the pump to be tested are separated from the other boilers and all cross connections of the water or steam pipes are broken and blanked off. Then no steam may be used for any other purpose and no water can be brought into the boilers, except by the one feed pump, which draws its supply from a tank, the water in which has been weighed.

Now if there is any water collected in a separator between the boilers and the engine, or used for any purpose whatsoever, other than for the engine, it must be carefully determined.

Suppose W pounds of water are pumped into the boiler, w pounds used in a calorimeter to determine the quality of the steam, w' pounds taken from the separator, and w'' pounds used in the jackets and reheaters. The amount of water used then by the main pumping engine is $W-w-w'$ pounds. Care must be exercised to investigate the piping to see that there is no

chance of leakage of the water or steam in any way, and all joints into other lines should be broken and blanked off. A leakage test of several hours' duration should be made at the end of the test, with the main engine shut off to determine the loss from the boilers and piping. The quantity so found is subtracted from W.

By taking readings of the steam gauge, barometer, and on the calorimeter, the absolute pressure and the quality of the steam can be determined, and hence the quantity of heat above 32° F. supplied per pound. The total heat supplied above 32° F. is then

$$H=(W-w-w')(q+xr)$$

where

q = heat of the liquid;
x = quality of steam;
r = heat of vaporization.

For superheated steam $q+xr$ becomes $q+r+\int c_{p}dt$.

In general, the steam jackets and reheaters drain through a trap into the boiler feed, and hence return some heat to the boiler. To determine this the weight of the water from them is found as before described and the temperature of that water before it is cooled. The temperature is found by placing a thermometer cup in the pipe and taking temperature readings on a thermometer placed therein. The heat then returned is

$$w''q_2''$$

The feed pump, under actual running conditions, takes water from the hot well, hence the temperature of this must be observed and the heat supplied by this source determined. If the temperature be t_3, the quantity returned is

$$(W-w-w'-w'')q_3$$

The total heat chargeable to the engine is

$$H=(W-w-w')(q+xr)-w''q_2''-(W-w-w'-w'')q_3,$$

then

$$\frac{\text{Work}}{H}\,1{,}000{,}000=\text{Duty}.$$

If the duty to be found is on the basis of 1,000 pounds of dry steam, the steam used must be expressed in terms of equivalent dry steam. The equivalent dry steam is given by

$$W_\varepsilon = \frac{(W-w-w')(q+xr-q_3)}{(q+r-q_3)}, \text{ if saturated,}$$

or

$$W_\varepsilon = \frac{(W-w-w')\left(q+r+\int_{t\,\text{sat.}}^{t\,\text{sup.}} c\,dt - q_3\right)}{(q+r-q_3)}, \text{ if superheated.}$$

$$\text{Then duty} = \frac{\text{Work}}{W_\varepsilon}\ 1{,}000.$$

There are other quantities which are reported. The mechanical efficiency of the pump is determined by taking indicator cards from the water and steam cylinders and determining the horse-power of each end, and the ratio of these gives the mechanical efficiency. The first subtracted from the second gives the frictional loss.

Call the horse-power of the steam end I.H.P. and of the water end D.H.P. Then mechanical efficiency $= \dfrac{\text{D.H.P.}}{\text{I.H.P.}}$.

$$\text{I.H.P.} - \text{D.H.P.} = \text{H.P. of engine friction.}$$

The quantity of steam used per hour, divided by the I.H.P., is the steam per I.H.P. hour, a result useful for comparison with other kinds of engines. This is

$$\frac{\dfrac{W-w-w'}{\text{time}}}{\text{I.H.P.}} = \text{Steam per I.H.P. hour.}$$

or better

$$\frac{\dfrac{W_\varepsilon}{\text{time}}}{\text{I.H.P.}} = \text{Dry steam per I.H.P. hour.}$$

The quality of the steam, or in other words the amount of condensation, in each cylinder at any time after cut-off in that cylinder can be determined, first, by finding the volume (including clearance volume) and pressure at that point from

the indicator cards, and second, from the table of the properties of steam, the volume occupied by one pound of steam at this pressure. From the quantity of steam used in the cylinder, the number of pounds per stroke is found; and on adding to this the steam in the clearance space the total steam in the cylinder may be ascertained. The clearance weight is approximated by considering the steam dry at compression and then finding weight by the formula:

$$Wt = \frac{\text{Volume}}{\text{Specific Volume}}.$$

The product of the specific volume and the total weight per stroke gives the volume which should be occupied. The ratio of the actual volume at the point to this amount represents the percentage steam present at that point, the remainder being water, the volume of which has been neglected. In this way the initial condensation may be determined.

The amount of heat absorbed by the surface condenser is $W'(q'-q'')$, where W' is the weight of the condensing water, and q' is the heat of the liquid leaving the condenser and q'' the heat of the liquid in the entering condensing water.

This quantity of heat removed by the condenser may be called H'; then

$$H - H' - \text{Work} = \text{Radiation},$$

or

$$100 \frac{\text{Radiation}}{H} = \% \text{ loss by radiation.}$$

The following precautions are to be noted in making observations:

First, carefully arrange observation sheets and become familiar with readings before beginning test.

Second, after noting observations make a check observation when possible.

Third, keep a vigilant watch for leaks of any kind.

Fourth, note everything in book that happens out of the ordinary.

Fifth, make note of number and position of all instruments used.

Sixth, calibrate and check all instruments and measurements.

Seventh, in weighing coal or water, take the weight of vessel empty and full, together with the time, in each case.

Eighth, note the time of all observations, except when several different ones are made in quick succession, then note the time of the first only, but take them always in the same order.

Ninth, always have instruments ready for next operation before leaving them.

The data and results called for in duty trials are given in Kent's Pocket-book, and the following additional results and data are also advisable for investigation of the pump action:

Plunger displacement per stroke.
Amount pumped, cubic feet.
Quantity pumped, computed from displacement.
Quantity of water used in condenser.
Quantity of heat absorbed by condenser.
Quantity of heat absorbed by condenser, per square foot of surface.
Percentage loss by radiation.

These tests are usually made to find whether or not the guaranteed duty is met, but an investigation in the same manner may be made to find whether or not a given device or arrangement is of value. For instance, in many cases it is found that the use of the reheating coils in the receivers are not a source of gain and at times a test without certain jackets will give better results than a test made with the jackets in service.

Superheated steam can be used and its effect determined in this manner. The use of sufficient superheat to cut down the initial condensation and give dry steam at cut-off will pay, but beyond this the expense in making the superheated steam will be more than the gain in the engine. The only reason why it should be of value when the initial condensation is eliminated, is the fact that when the steam is dry or superheated the walls

do not take up as much heat as when covered with a film of moisture.

One reason why superheated steam should not add to the efficiency, the amount apparently shown by the increase of the upper temperature T_1 of the expression for the Carnot efficiency,

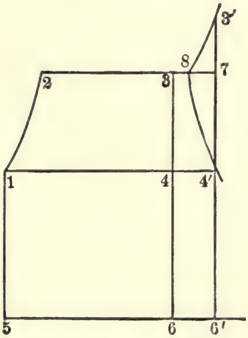


FIG. 336.—Entropy Diagram.

$$E = \frac{T_1 - T_2}{T_1},$$

is the fact that this heat is not supplied at a fixed temperature T_1, but at a gradually increasing temperature, and hence although the efficiency of the cycle is increased a slight amount the increase is very small. This may be shown by Fig. 336, which is a $T\phi$ diagram. In the figure, 1, 2, 3, 4 represents the cycle of an engine using wet steam, while 1, 2', 3', 4' is one for an engine supplied with superheated steam. The efficiencies are

$$\frac{1, 2, 3, 4}{5, 1, 2, 3, 6} = E_{\text{sat.}},$$

and

$$E_{\text{sup.}} = \frac{1, 2, 3', 4'}{5, 1, 2, 3', 6'} = \frac{1, 2, 7, 4' + 8, 3', 7}{5, 1, 2, 7, 6' + 8, 3', 7} > \frac{1, 2, 7, 4'}{5, 1, 2, 7, 6'},$$

$\frac{1, 2, 1, 4'}{5, 1, 2, 7, 6}$ is slightly greater than $\frac{1, 2, 3, 4}{5, 1, 2, 3, 6}$ so that $E_{\text{sat.}}$ and $E_{\text{sup.}}$ are almost the same.

The use of jackets on the cylinder heads and barrels is recommended although tests do not always show an increase of economy.

The overall efficiency of the engine is given by dividing the duty per 1,000,000, B.T.U. by 778,000,000, which is the number of foot pounds equivalent to 1,000,000 B.T.U.; this number has reached a value of 0.23. This means that of the heat supplied to the test pumps 23 per cent is utilized.

The preparation of a pump for a duty test consists in much adjustment of valve-gearing and other steam apparatus which consumes time. Many preliminary runs are made to discover the effect of changes and this can be done only by a systematic method of keeping data and making alterations. One change should be made at a time and its effect determined before proceeding to another change. In all of this work a positive circulation must be maintained in the heating coils and jackets, for if the water of condensation is not removed these cease to operate properly.

To illustrate the data and results of a duty test, the following has been taken from a test on a 20,000,000 gallon pump at the Lardner's Point Pumping Station in Philadelphia, conducted by Francis Head, mechanical engineer of the Bureau of Water of the city of Philadelphia, and Edgar G. Hill, mechanical engineer representing the Holly Manufacturing Company.

"At Lardner's Point are three pumping stations, No. 1, No. 2 and No. 3. Station No. 1 was formerly known as Frankford station, and contains boilers and pumping engines that have been in service a great many years, which are now retained only as a reserve, the machinery being entirely too expensive to keep in regular service.

"Stations No. 2 and No. 3 are new pumping stations throughout; the buildings and equipment therein were constructed for the new filtration distribution system. Each consists of an engine house and boiler house. The two stations are placed contiguous and for all practical purposes are one station.

"In engine houses No. 2 and No. 3 are twelve vertical triple expansion self-contained pumping engines of 20,000,000 U. S. gallons daily capacity each, designed for a normal head of 225 feet, but capable of operating economically against heads ranging from 180 to 280 feet.

"These twelve pumping engines, each substantially a duplicate of the other, designed, constructed, and installed by the Holly Manufacturing Company of Buffalo, N. Y., make an installation the largest and most comprehensive of its type, not only in the United States but in the whole world.

"The water which these engines pump is supplied from the filtration plant of the city of Philadelphia at Torresdale, between two and three miles from Lardner's Point pumping station, through an underground conduit leading from the Torresdale filtration plant to the Lardner's Point pumping station. It is pumped from the Lardner's Point pumping station through a number of delivery mains, ranging in diameter from 48 to 60 inches, to various parts of the city of Philadelphia.

"Ordinarly, eight or nine of these engines discharge the water through two 60 inch delivery mains, which are so connected together as to form substantially one main of twice the capacity of a 60 inch pipe, against a head of approximately 185 feet. The other three or four engines ordinarily pump through a separate pipe system against a head of approximately 275 feet. To show the action of the governor the following is mentioned:

"At 8 o'clock, the morning of December 10, 1909, at a point about one thousand feet from the pumping station, a 12 foot length of the 60 inch delivery main, into which eight of these pumping engines were at that time delivering water at the rate of over 160,000,000 gallons per day, against a head of approximately 185 feet, split from end to end, instantaneously reducing the work from full load to almost no load. Five of these pumping engines were in Station No. 2, and three in Station No. 3.

"All of the pumping engines were under such perfect control of the governors and automatic safety devices that only one of the eight attained sufficient speed to make the automatic shutting-down device operative, the result being that one engine stopped automatically, and the other seven ran at a uniform speed until gradulaly closed down by the employees at the station. Absolutely no damage was done to any of the engines or to any of the machinery in the pumping stations.

"The broken 60 inch delivery main was promptly repaired and the engines were all in service again the afternoon of the same day the accident occurred.

"This is undoubtedly the first time the governing apparatus of so many pumping engines has been subjected simultaneously to a test of this character. That none failed to work properly

and satisfactorily, speaks louder than words for the reliability of the governing apparatus.

"One of the twelve engines was officially tested March 9–10, 1910, and developed a new high duty record for pumping engines fitted with attached jet condenser, on a 24–hour trial.

"During the test, readings of the water pressure were taken every five minutes from a correct pressure gauge checked by a mercury column. A complete round of observations was taken every fifteen minutes. All readings and weights were checked by two observers, one representing the city and one the company.

"An eight-hour boiler leakage test was made before the duty trial, also immediately after the duty trial.

"A summary of the principal results is given below, by the makers:

Size of engine		30″, 60″, 90″x33″x66″
Total water pressure, lbs.		95.74
Average total feed water per hour, lbs.		9,265.625
Feed water corrections, average per hour, lbs.:		
Drips from steam header and steam separator	135.27	
Boiler leakage, average of two tests	196.39	
Total		331.66
Average total steam per hour delivered to engine, lbs.		8,933.97
Entrainment, per cent		1.135
Average total dry steam per hour delivered to engine		8,832.56
Water pumped per 24 hours, gallons		21,218,788
Duty per 1,000 lbs. steam delivered to engine		182,382,200
Duty per 1,000 lbs. dry steam		184,476,200"

In the specifications for pumping engines definite statements should be made of the manner of conducting the test and of the working out the results. As an example, the following quotations are made of clauses from the specifications for the Lardner's Point pump:

"Section 119. *Duty Test.* During the period of probation and before its final acceptance, each engine shall be subjected to a duty test of twenty-four (24) hours' duration. The test shall be conducted by two (2) engineers, one (1) to be selected by the director, and the other by the contractor, etc.

"Section 120. *Determination of Head.* The nead for computation of duty will be the sum of the head in feet indicated on a gauge attached to the discharge main beyond the last pump, and the vertical elevation of the center of this gauge above the level of water in the pump well. The level of water in the pump well shall be determined by a suitable float gauge, and no correction or allowance will be made for any friction losses between the water in the pump well and in the discharge pipe just beyond the pump.

"Section 121. *Total Head.* The total head for the contract duty test shall be not less than two hundred and fifteen (215) and not more than two hundred and twenty-five (225) feet.

"Section 122. *Capacity.* The capacity of the engines shall be in the duty test not less than twenty million (20,000,000) gallons per twenty-four (24) hours, at a speed of not more than twenty (20) revolutions per minute. The capacity of the pumps during the duty tests will be determined by plunger displacement, and no correction will be made for slip unless leakage from plungers and valves is found by test to exceed two and one-half ($2\frac{1}{2}$) per cent.

"If the leakage or slip is found by Pitot meter to be more than two and one-half ($2\frac{1}{2}$) per cent, the capacity and duty shall be computed from the measured capacity.

"Section 123. *Stuffing Box Leakage.* The leakage from all the water stuffing boxes of the engine shall not exceed five hundred (500) gallons per hour. This shall not be charged against the engine.

"Section 124. *Feed Pumps.* The direct-connected feed pumps shall be operated during the duty trial, supplying water to the measuring tanks, and running against the usual discharge pressure. One auxiliary feed pump shall be operated to remove condensation. No allowance will be made for these pumps.

"Section 125. *Steam Measurement.* The water fed to the boilers shall be weighed and any condensation in the lines and drains from the live steam separator shall be deducted. All steam passing through the separator to the throttle valve shall be charged against the engine as dry steam.

"Section 37. *Uniformity of Steam Diagrams.* The construction and adjustment of the pump valves and steam valve gear, and the balancing of the plungers and other moving parts, shall be such as to give approximately uniform engine indicator diagrams from each of the steam cylinders on the up and down strokes."

In making the report on the test of this pump, the following statements are made:

"The level of water in the pump well was determined by a gauge glass placed near the pump suction instead of a float, as this gave better results.

"Under Section 122, regarding correction for slip, as before this was assumed to have been under $2\frac{1}{2}$ per cent, and no correction was made for it.

"Under Section 123, stuffing box leakage was measured and found to be less than allowed.

"Under Section 124, which calls for the operation of one auxiliary feed pump to remove condensation, the results of the test made on July 20th, 1909, reported in the test of No. 16, were used, and the amount so determined charged against the engine.

"Under Section 125, the same six inch steam line was used to convey steam from boiler No. 26 to the pump. As before, the drip was trapped, condensed, weighed and deducted from the feed water.

"The correctness of the feed water scales was checked by Fairbanks Standard weights, and the steam and water gauges were corrected by a Crosby dead weight tester, which had been carefully calibrated.

"The water pressure gauge was further checked by a mercury column which was set up beside the pressure gauge. The density of the mercury used was determined in the testing laboratory of the Department of Public Works.

"On March 8th, the suction and discharge valves were tested and made tight.

"On March 8th, a leakage test of the boiler and piping was made from 9.47 A.M. to 5.47 P.M. On March 10th, another

test was made from 1.00 to 7.55 P.M. The average net loss was 196.392 pounds per hour.

"The capacity of the engine, as actually run during the test, from plunger displacement, was 21,218,481 gallons per 24 hours at a speed of 20.13125 R.P.M.

"The engine worked smoothly and satisfactorily during the entire test, and the requirements of the specifications as to economy and capacity on the duty test have been fully complied with."

The following data was obtained during test from average readings:

Data and Results

Engine tested, contractor's number	604
Water Bureau number	14
Date of test	March 8, 9, and 10, 1910
Duration of test	24 hrs.

Capacity

Revolutions during test	28,989
Average revolutions per minute	20.13125
Average diameter of plungers, inches	32.985
Average stroke, feet	5.4961
Number of plungers	3
Displacement per revolution, gallons	731.9494
Displacement per 24 hours, gallons	21,218,481
Displacement per 24 hours, at contract speed	21,080,144
Water used to lubricate plungers, per hour, gallons	360

Work Done

Head pumped against—	
Pressure, corrected gauge	86.918 lbs. = 200.50 ft.
Suction lift to center of pressure gauge	8.822 lbs. = 20.35 ft.
Total	95.740 lbs. = 220.85 ft.
Work done per hour	1,629,397,423 ft.lbs.

Duty

$$\text{Duty} = \frac{\text{Foot-lbs. per hr.} \times 1000}{\text{Net steam charged to pump per hour}}$$

$$= \frac{1,629,397,423,000}{8991.713}$$

$$= 181,211,013 \text{ ft.lbs.}$$

This is slightly different from the builders' computations given above.

Pressures

Throttle gauge reading, lbs. per sq.in	190.82
Corrected for height	180.19
First receiver, lbs. per sq.in	33.8
Corrected for height	25.4
Second receiver, inches vacuum	9.5
Vacuum in condenser (reading of mercury column) inch	27.68
Average barometer at 32°F. sea-level	14.789 lbs. = 30.06
Average barometer at floor level and room temperature	14.868 lbs. = 30.20

Jacket and Receiver Data

High-pressure jacket, lbs. per sq.in	180.19
Intermediate jacket, lbs. per sq.in	34.5
Low-pressure jacket, lbs. per sq.in	0.5
First receiver drip, per hour	422.16
Second receiver drip, per hour	360.87
Jacket drip, per hour	660.62
Total drip, per hour	1443.65
Per cent of steam passing throttle	16.16

Temperatures

In exhaust pipe, 4 ft. below L.P. cylinder	113° F.
Water pumped	40°
Condensing water	40°
Water leaving condenser	70°
Water after passing feed heater in exhaust pipe	85°
Air in engine room at mercury column	80°
Outside average	30°

Calorimeter

Taken after test, instrument between throttle and cylinder:

Pressure, lbs	176.7
Temperature	297.25° F.
Moisture, per cent	1.133

Evaporation

Water pumped to boiler per hour, lbs	9265.625
Boiler leakage per hour, lbs	−196.392
Water evaporated per hour	9069.233
Engine room drip per hour	−135.270
	8933.963
Steam required to run drip pump per hour	+57.750
Net steam charged to engine per hour	8991.713
Boiler horse-power developed	324
Boiler rated horse-power	500

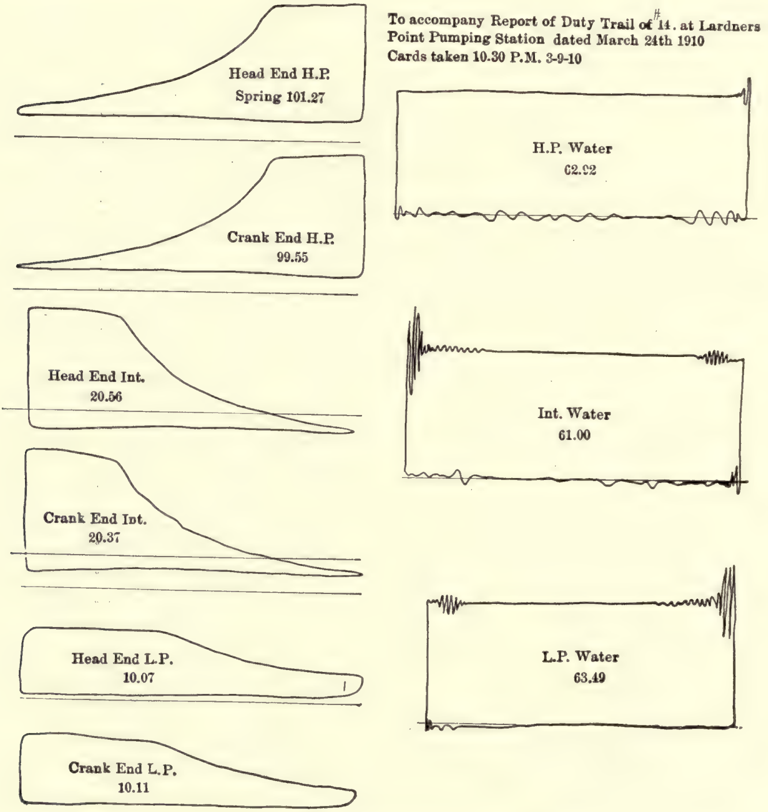


FIG. 337.—Indicator Cards from Lardner's Point.

"The boiler used was an Edgemoor water tube boiler No. 26 of the same size and type as used on the previous tests.

"It was connected to the engine by a special six inch steam pipe which was well covered.

"Indicator cards are arranged to correspond to the position of the cylinders from which they were taken. These are shown in Fig. 327.

"The distribution of power between the different cylinders as determined from the cards is shown on the following table:

Steam.	Head.	Crank.	Total.	Per cent of Total.
High-pressure cylinder..	163.47	168.55	332.02	38.6
Intermediate cylinder...	135.35	133.75	269.10	31.3
Low-pressure cylinder...	130.10	130.12	260.22	31.1
	428.92	432.42	861.34	100.0

Mechanical efficiency, ratio of net delivered water horse-power to indicated steam horse-power..................	95.54
Horse-power from volume and head of water................	822.93
Water horse-power indicated...............................	839.8
Excess of indicated over delivered water H.P...............	2.1%
Steam passing throttle per indicated H.P. per hour, lbs...	10.37
Steam passing throttle per delivered water H.P. per hour, lbs.	10.86
Heat units from steam pressure to vacuum temperature used per I.H.P. per minute..............................	193.04
Heat units passing throttle from steam pressure to vacuum temperature used per delivered water H.P., per minute.	202.05
Efficiency from heat in steam delivered to engine above temperature in exhaust pipe to work done in discharge main.	20.99

NOTE.—In the test the reheating coils in both receivers were out of service."

This test represents one of the highest duties obtained on pumping engines and should be studied, as many results can be obtained from the data.

CHAPTER X

HIGH DUTY PUMPS AND WATER WORKS STATIONS

To study the modern form of the water works high duty pump and its installation a number of examples will be given.

The Western Pumping Station of the Water Works of Cincinnati, Ohio, is shown in the Figs. 338 to 341. The station at present contains three 25,000,000 gallon pumps for low service and three 12,000,000 gallon pumps for high service, built by the Holly Manufacturing Co. of Buffalo, N. Y. The station, Fig. 339, contains space for an additional pump of each kind. The water enters through a gravity tunnel to a well, and from this point a conduit of pipe extends to the suction of the various units. Two suction lines are taken off the main conduit at each pump, one of them passing through the condenser, and these enter the suction valve chambers, as will be seen later, and finally connect together in a suction air chamber at the end. The two discharges from each side of the pump are connected into the discharge main, which is connected to two service mains leaving the station on each side of the center.

The branches from the sides of the pump may be cut off by valves which are shown by the conventional cross, and by valves in the service lines; any of these may be cut off from the station. Valves in the main discharge line of the station make it possible to use certain of the pumps on one service main. Such arrangements are valuable at times for service and for experimental work. Attention is called to the fittings left for the installation of the new pumps.

The steam is generated in the boiler room in which the boilers are placed on the side away from the pump room. This makes a longer steam line, but with it the chimneys can be located in a better manner. The open space in front of the

boilers is so wide that tubes may be removed from the boilers. This space forms a convenient place for firing. The boilers are

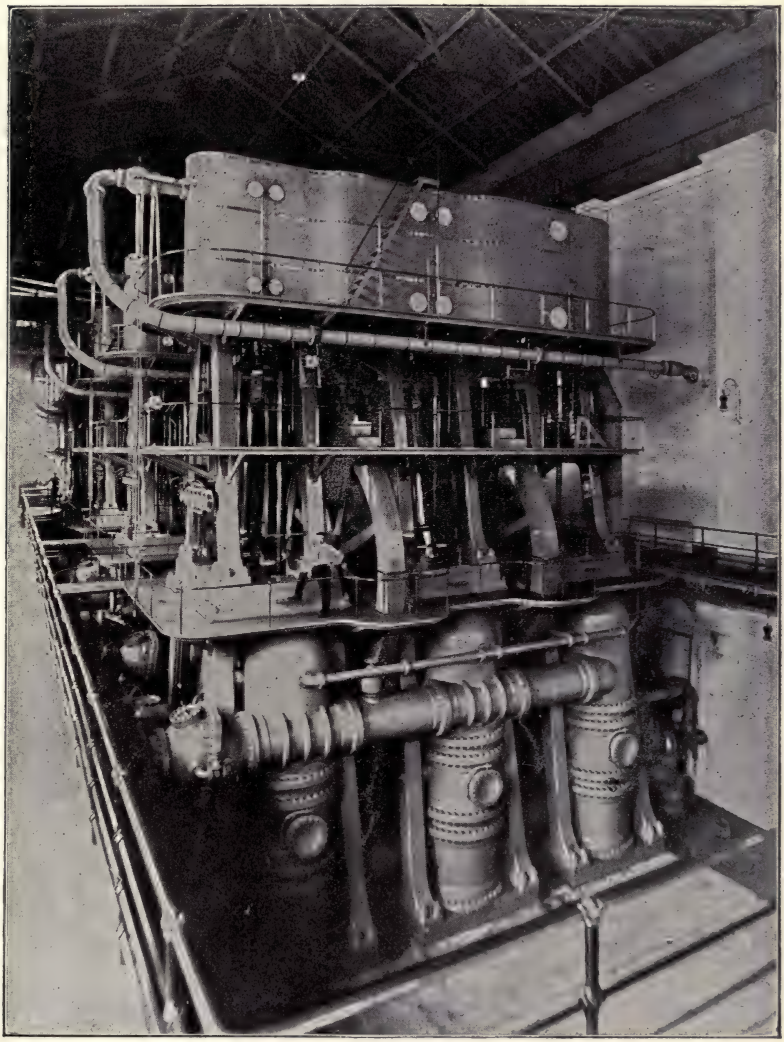


FIG. 338.—Western Pumping Station

connected to a steam header carried on the wall between the boiler room and pump room and branches lead to the engine.

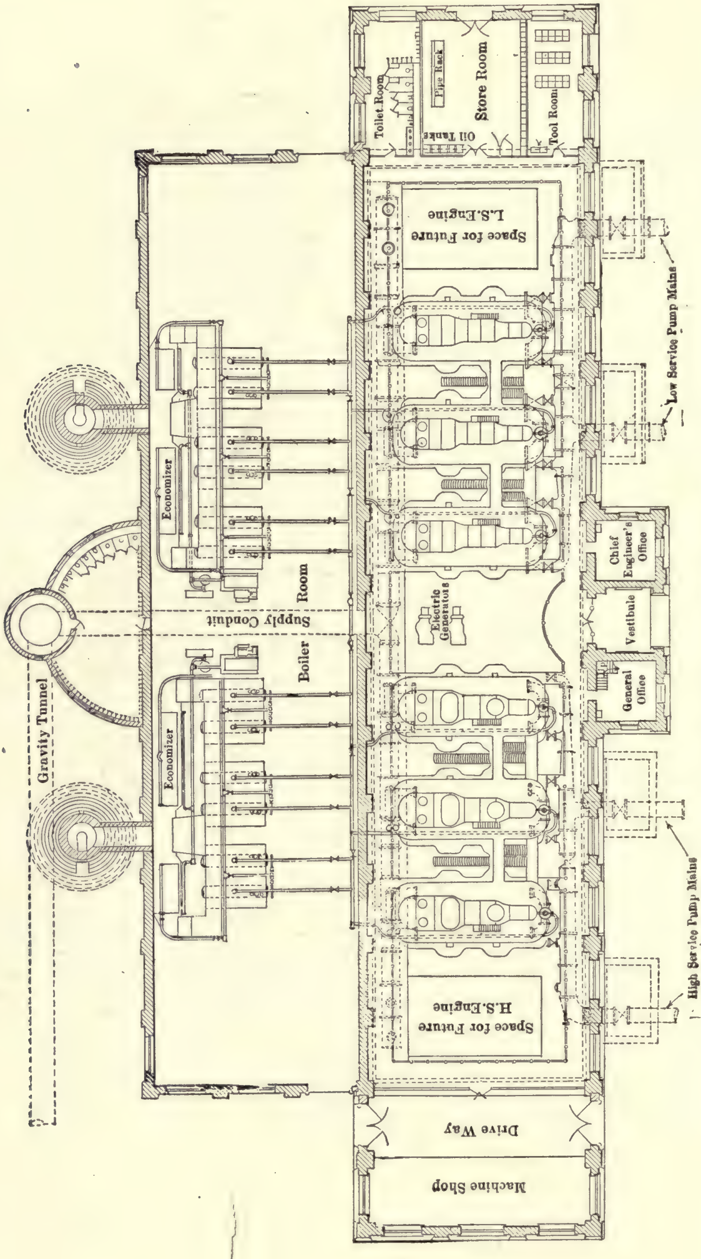
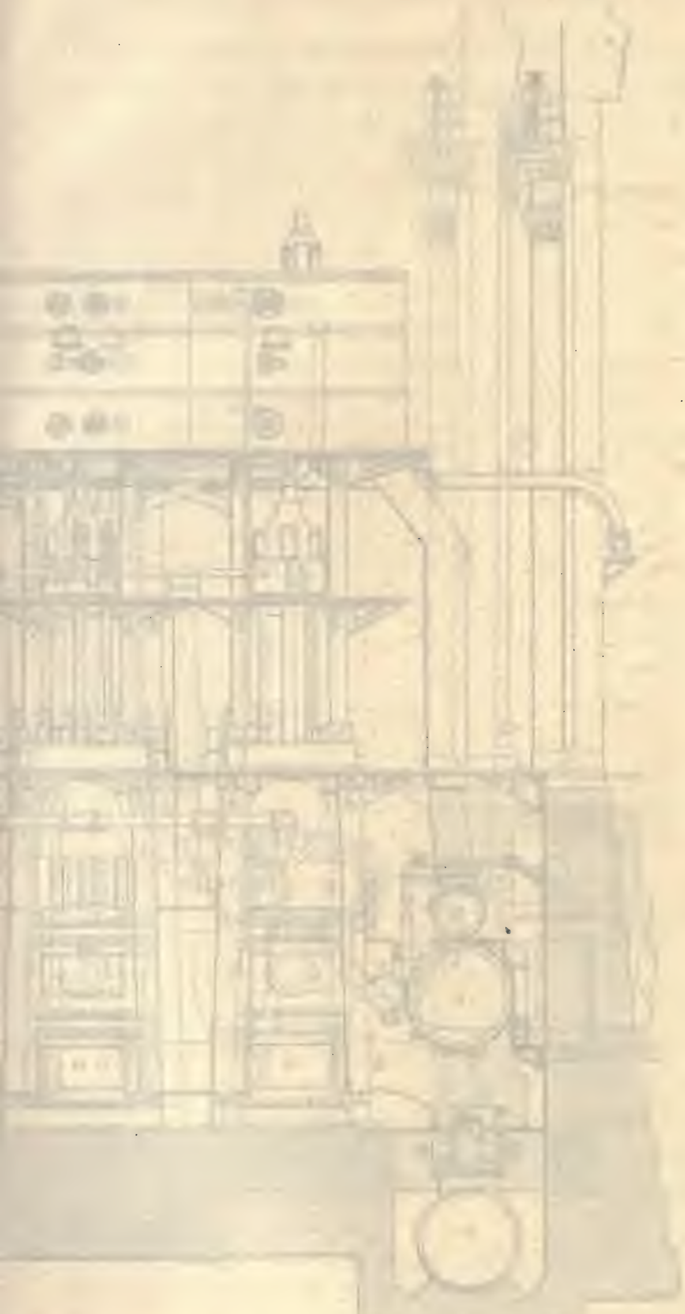


FIG. 339.—Cincinnati Station.



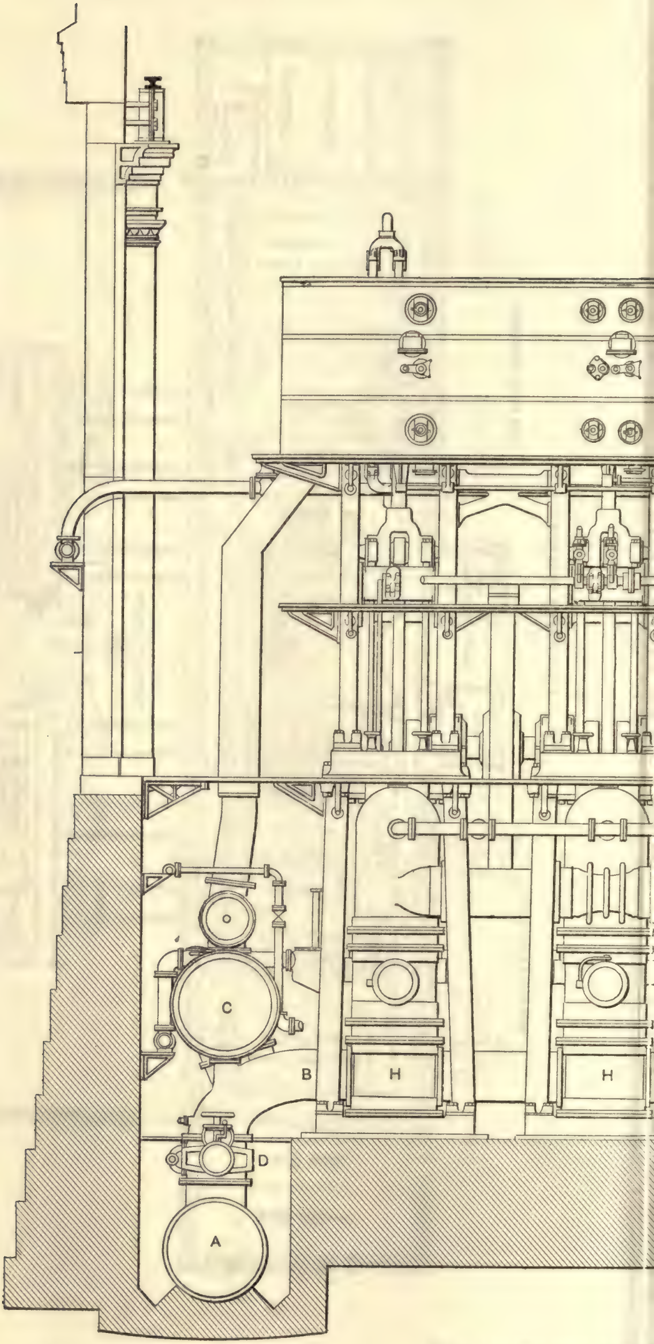
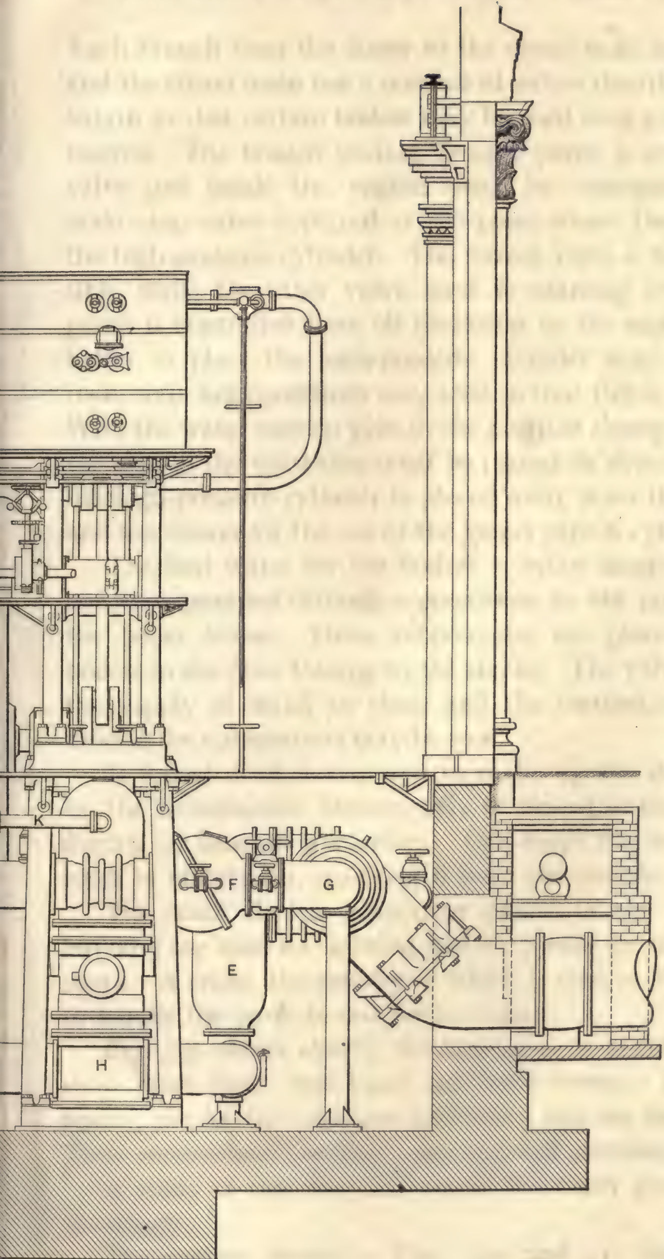


FIG. 340.—Front Elevat



Holly Cincinnati Pump.

(To face page 431)

Each branch from the boiler to the steam main has two valves, and the steam main has a number of valves distributed along its length so that certain boilers may be used with a given pump if desired. The branch leading to each pump is controlled by a valve just inside the engine room for emergency while the main stop valve is placed at the point where the steam enters the high pressure cylinder. The former valve is not very accessible, while the other valve used in starting or stopping the pump is controlled from all platforms or the main floor. It is better to place the high-pressure cylinder next to the boiler room wall, but conditions may arise so that this is not advisable. With the water suction pipe in the position shown in the plan of the station the condenser must be placed on this end, and hence the high-pressure cylinder is placed away from the boiler room and the reason for the use of the longer pipe is evident.

The feed water for the boilers is taken from the condenser and then pumped through economizers by the pumps shown in the boiler house. These economizers are placed behind the boilers in the flues leading to the stacks. The valves controlling the supply of water to these and the method of discharging around the economizers may be seen.

If forced draft is required to make up the draft reduction by the economizers, blowers may be placed in the boiler room, forcing air beneath the boilers. This keeps the air in the boiler room in circulation, renewing if from the outside.

The small electric generators placed in the center of the building are used for lighting and for power for the cranes and shops. A crane, the runway of which is seen in Fig. 338, serves to handle the parts in erection or repair.

Fig. 339 shows clearly the arrangement of offices, machine shop, store room, tool room and toilet rooms. There are two rooms, one for the engineers and oilers and one for the firemen. These rooms should be large, airy and well ventilated.

A study of this plan will reveal the many good features of the design.

The pumps shown in Figs. 340 and 341 for this station illustrate a type of high duty pump.

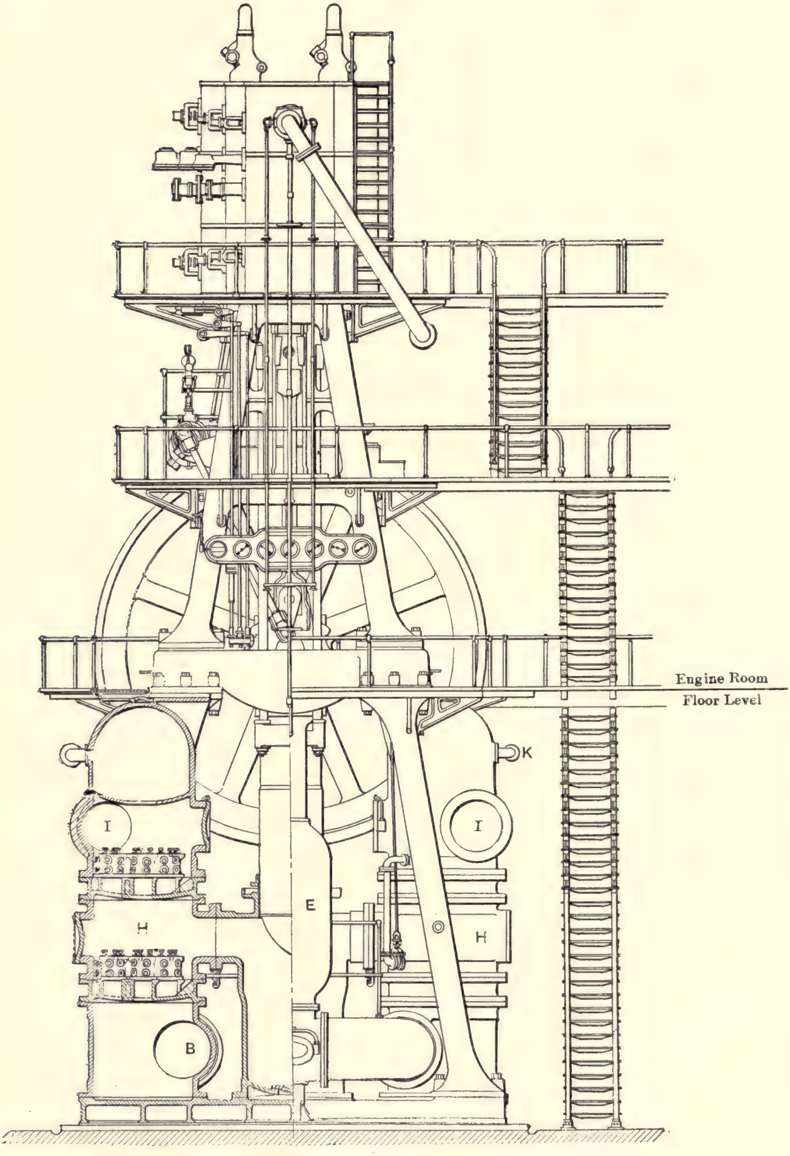


FIG. 341.—Holly Cincinnati Pump.

Water enters from *A*, passing the valves *D* on each side of the pump and entering the pipes *B*, which are continued through

the valve chambers *HH* to the end of the pump, where they combine and enter the suction air chamber *E*. The water is forced out by the plungers, as described on page 286, and is passed through the pipe *I* which is made a portion of the upper part of the valve chamber. The water from each side passes through check valves *F* and finally enters the discharge pipe at *G*. The discharge check valve is provided with a small by-pass valve for priming the pump.

The tops of the valve boxes form air chambers and these are supplied with compressed air when necessary through the pipe *K*. The steam end is triple expansion, with the three cylinders arranged in succession. The cross heads are equipped with four rods extending over the crank and shaft to the plunger cross head below. The end cranks are overhung but carry return cranks which are connected to the end cranks on the shaft *M*. These return cranks are so placed on the pins that they turn the shaft *M* positively, that is, when the cranks on one end are passing a dead point for the connecting rod, the others are at maximum throw. The shaft *M* is used to operate eccentrics, the motions of which serve to operate the Corliss valves of the cylinders.

In high duty engines the cylinders are usually jacketed and in many cases the receivers between the cylinders are heated by coils. Fig. 342 shows the internal arrangement of a reheating receiver. The coil is drained by means of a trap and this must act positively.

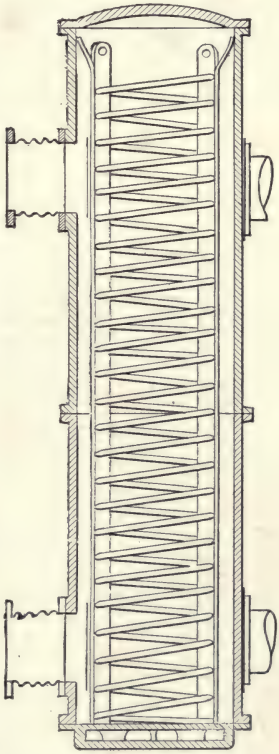


FIG. 342.—Reheater Receiver.

Fig. 343 is the interior of the Central Park Avenue Pumping Station in Chicago, which is similar to the Springfield Avenue

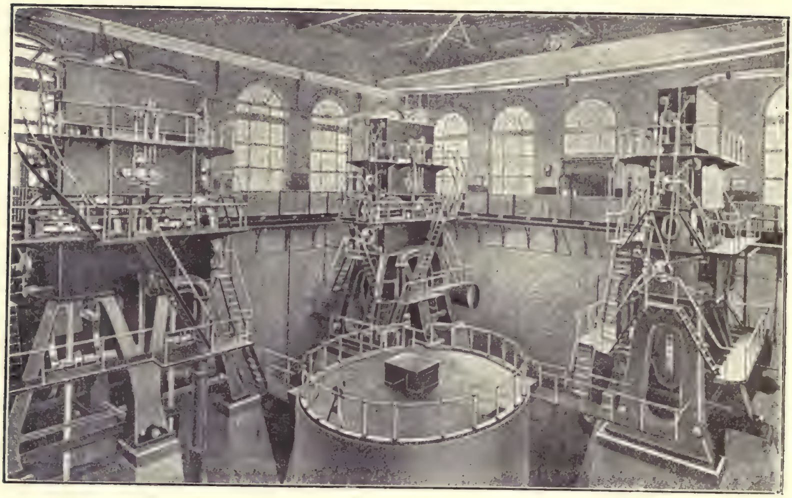


FIG. 343.—Central Park Avenue Station.

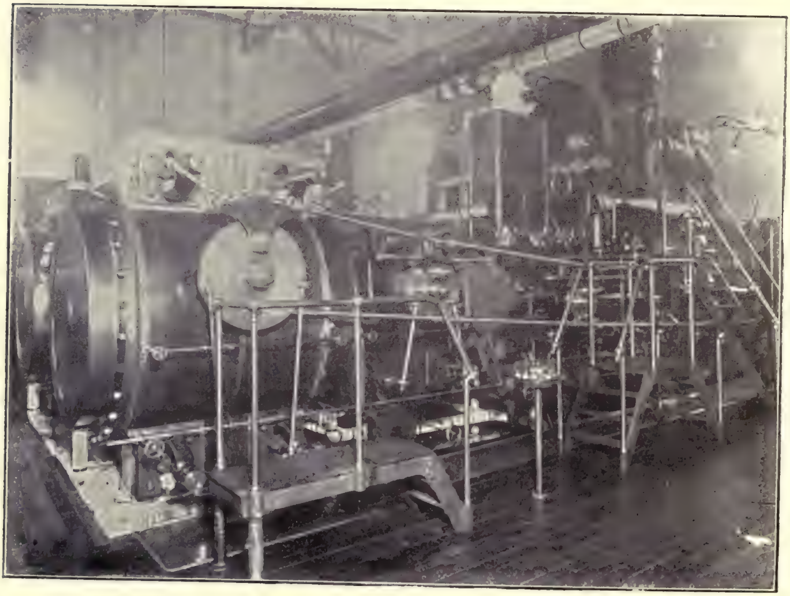


FIG. 345.—Worthington Pumps at Fall River, Mass.

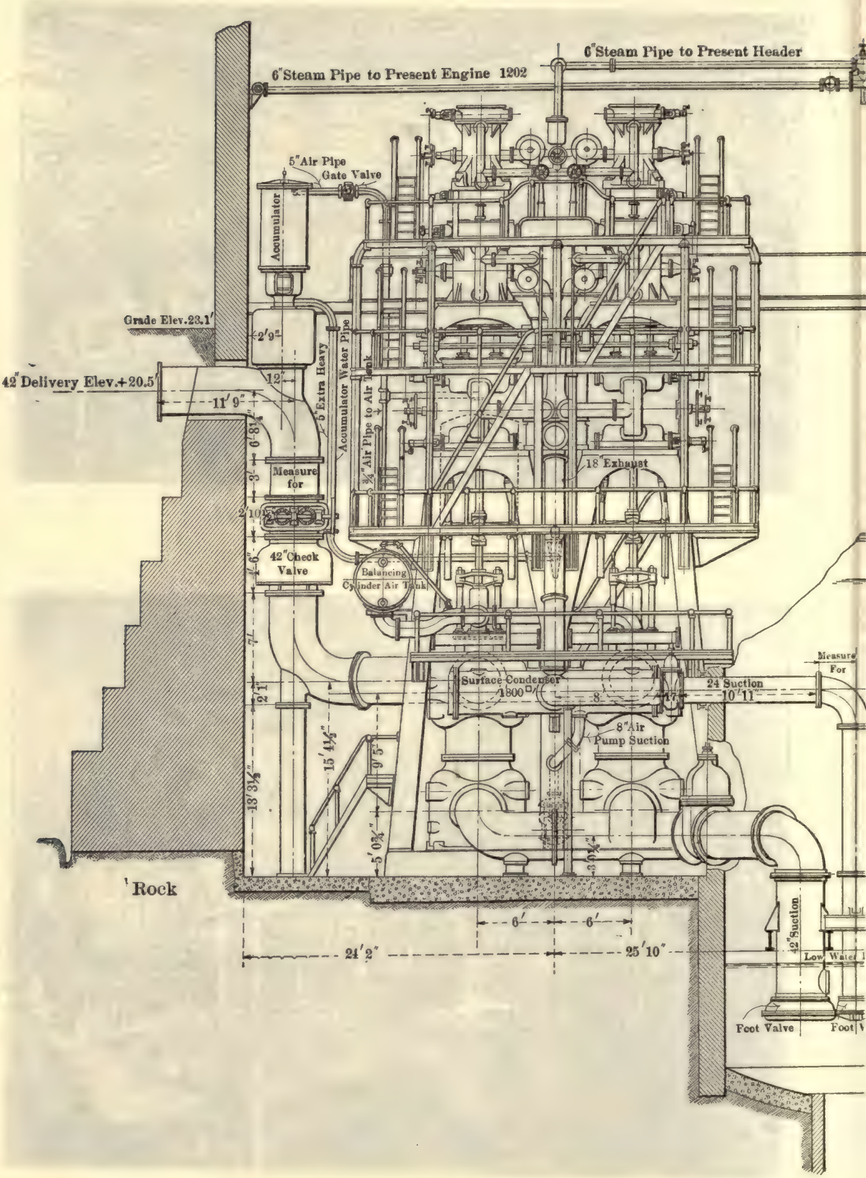
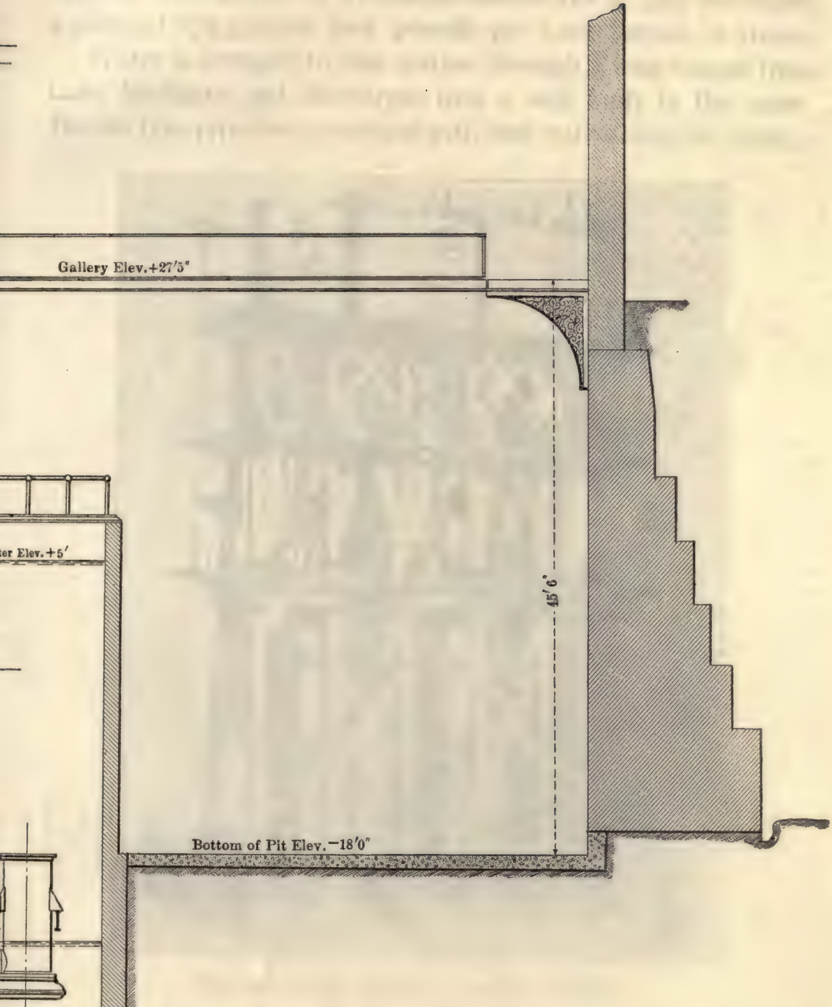


FIG. 344.—Worth



igh Duty Pump.

(*To face page 435*)

station. The pumps at the latter, shown in Fig. 344, developed a duty of 174,735,801 foot pounds per 1,000 pounds of steam.

Water is brought to this station through a long tunnel from Lake Michigan and discharges into a well built in the room. The suction pipes are provided with foot valves and the surface

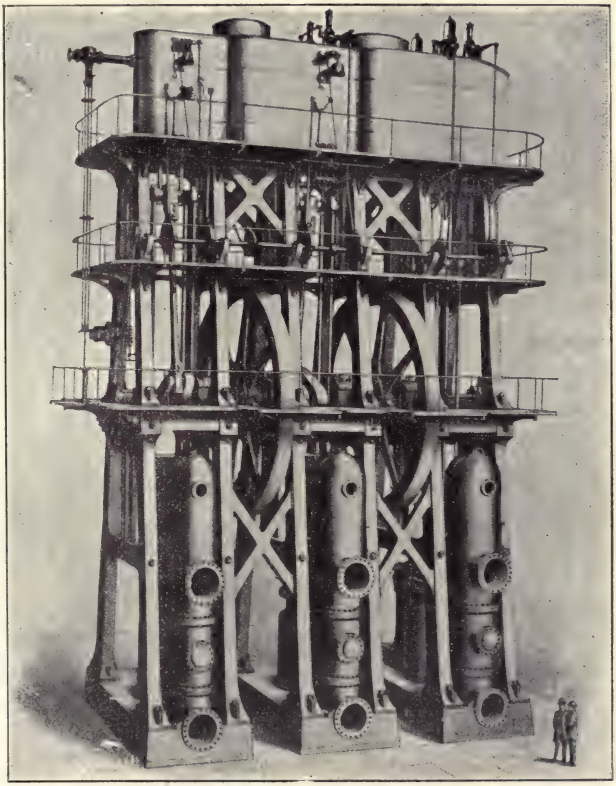


Fig. 346.—Allis Pumps at the Baden Station.

condenser uses water which is taken into the suction for condensing water. The pump, as described earlier, is a duplex, triple expansion, direct-acting pump, supplied with compensators. The cylinders are all in tandem and the weight is carried by a balancing piston so connected to the discharge main, as shown in Fig. 344, that should a break occur in the main the pump will come to rest. These pumps may be placed in a small floor area.

Fig. 345 shows a station equipped with a Worthington horizontal, triple-expansion, direct-acting engine. This pump was described on page 100.

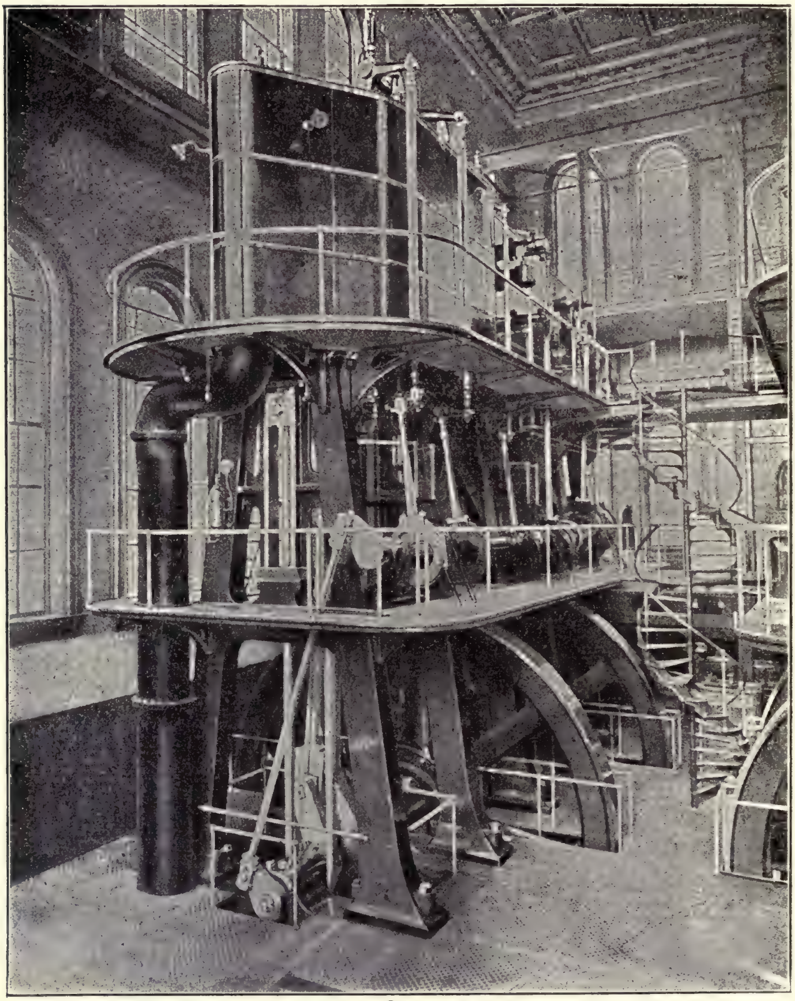


FIG. 347.—Allis Pumps at Bissel's Point.

Fig. 346 shows the appearance of the Allis-Chalmers pump, installed at the Baden station of the St. Louis water works, while Fig. 347 shows their pump at the Bissel's Point station

of the same system. In Fig. 347 the return crank driving the valve shaft on the first balcony is clearly seen. The valves of both these pumps are of the Corliss type on the high-pressure

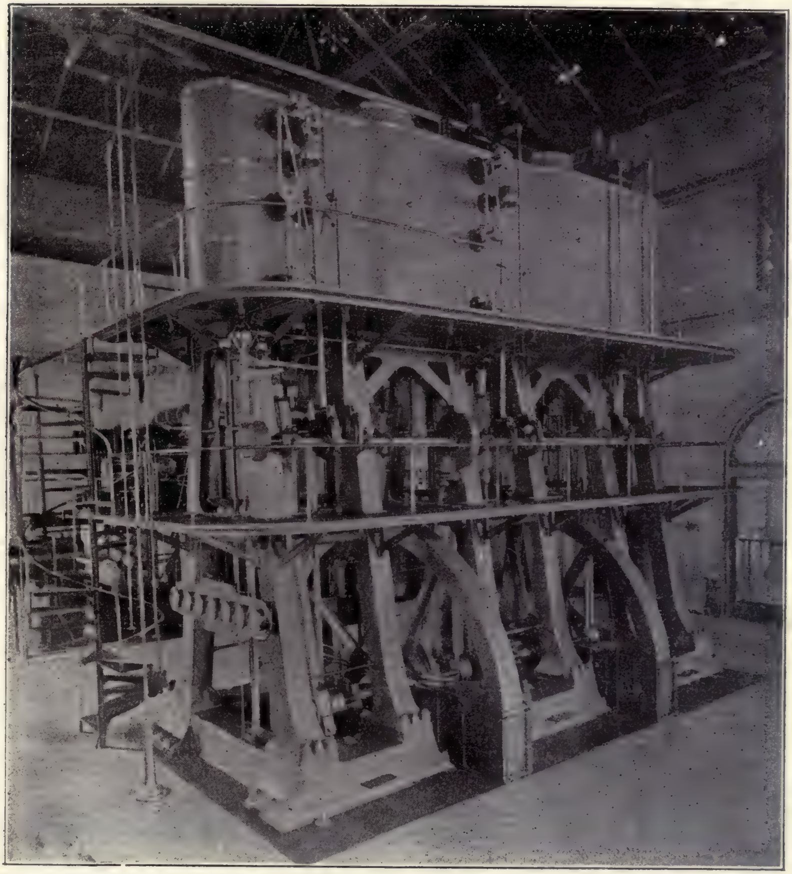


FIG. 348.—Holly Pump at Boston.

and intermediate-pressure cylinders, but on the low-pressure cylinder poppet valves are used.

Fig. 348 shows a pump built by the Holly Company for the Spot Pond Station of the Boston water works. In this pump the valve or eccentric shaft is driven by bevel gears. The exhaust valve of the intermediate-pressure cylinder of this pump and

both valves of the low-pressure cylinder are of the poppet type. The figure shows the various gauges and valves as well as the governor which actuates the cut-off on the high-pressure and

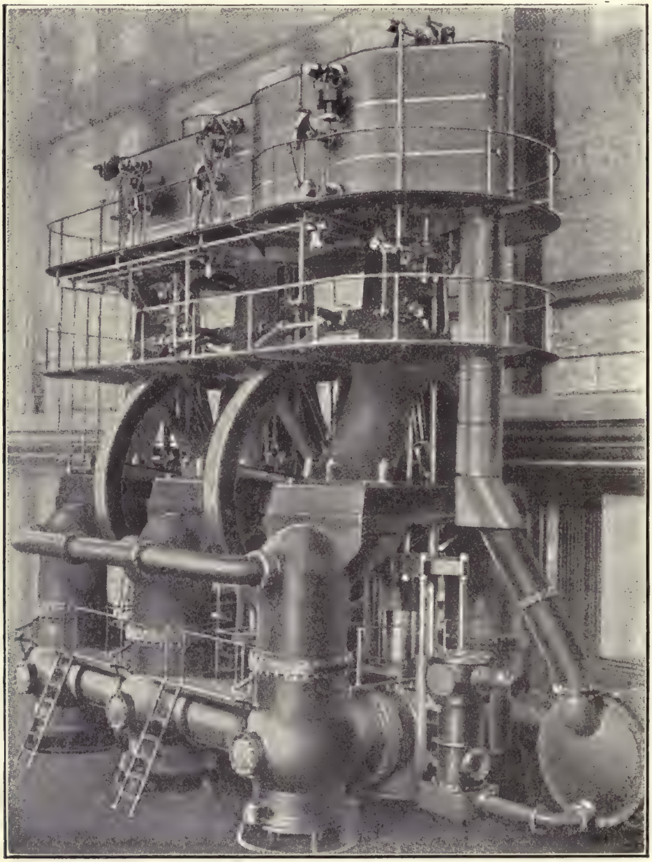


FIG. 349.—Allis-Chalmers Pump.

intermediate-pressure cylinders. In all these pumps the bracing of the frame is to be noted.

Fig. 346 shows clearly the type of complete frame often used with these pumps and in them the valve chambers are separate from the frame. In Fig. 349, showing an Allis-Chalmers pump, the valve chambers are used for supports. In the Holly

pump of Fig. 340, the separate air chambers are used, the bed plate at the floor line being carried by the inclined frames.

A small 2,500,000 gallon pump built by the Holly Manu-

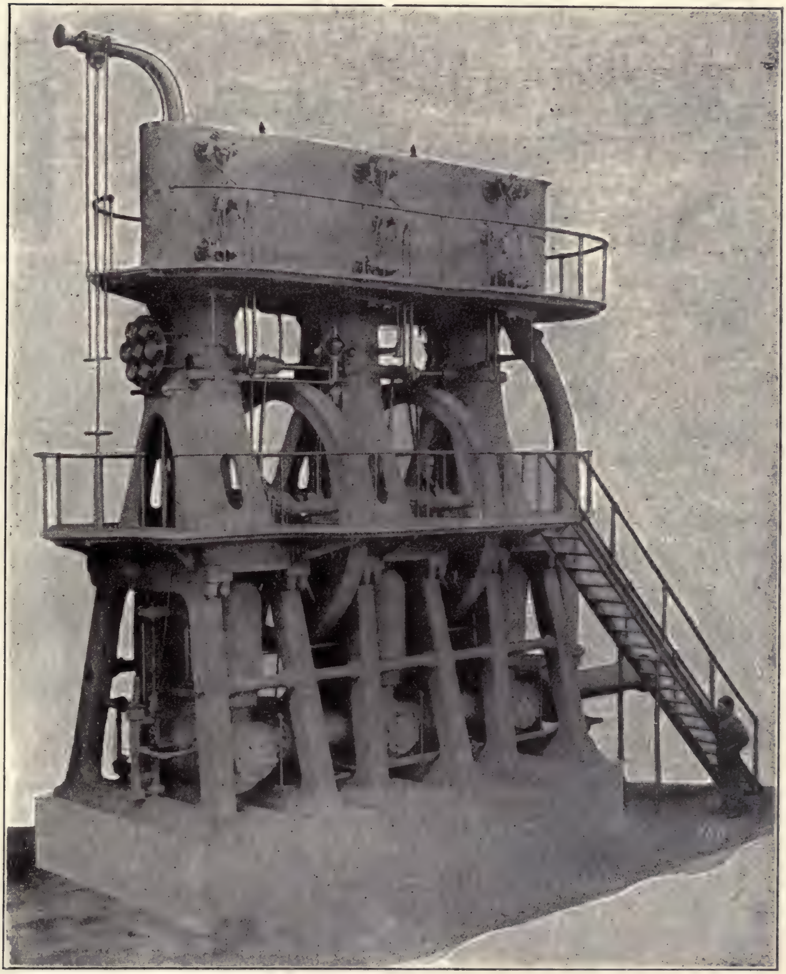


Fig. 350.—Holly Pump at Washington, D. C.

facturing Company for the Trumbull Street Pumping Station of Washington, D. C., is shown in Fig. 350. This pump gave a duty of 164,644,000 foot-pounds per 1,000 pounds of dry steam.

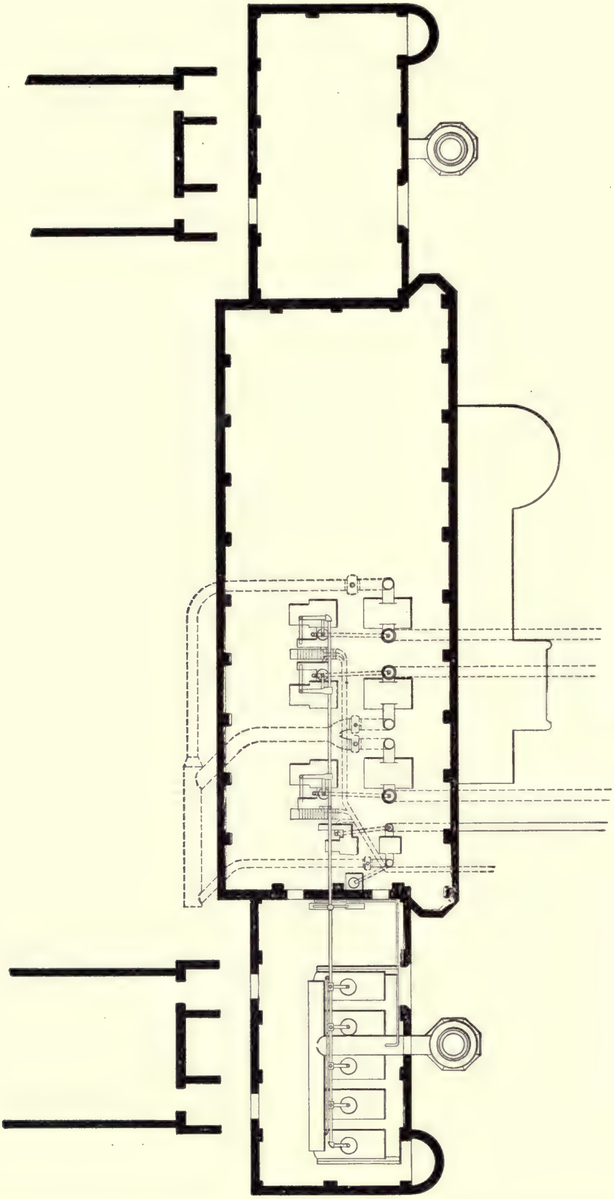


FIG. 351.—Brooklyn Station.

The valve shaft in this engine is oscillated by an eccentric on the main shaft. A governor controls the cut-off on the high-pressure

cylinder, but the cut off on the other cylinders is controlled by small hand wheels seen beneath the upper platform. All of the valves are of the Corliss type.

Fig. 351 represents the arrangement of a station in Brooklyn, N. Y. The boiler houses are placed at the ends of the pump house and the steam is taken to the engines by the main steam supply. The water ends are connected to a suction main on one side of the building and discharges are carried from each pump separately. In the rooms on the side of the pump house, as indicated by outline only, are placed the offices, shops, toilets and lockers, while the houses beside the boiler rooms are intended for coal storage.

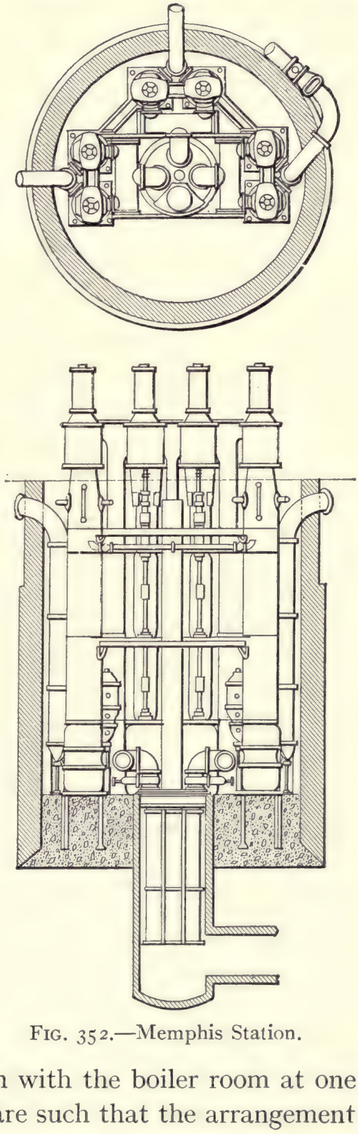


FIG. 352.—Memphis Station.

The storage of coal is important in all plants for the supply of water or for any other public service. This matter cannot receive too much attention. A supply for a week's consumption should be provided for and if conditions of coal delivery are poor this should be increased. In some cases a month's supply is none too much. The coal should be stored beneath cover as the alternate wetting and drying when uncovered causes the coal to deteriorate. With soft coal the pile should be ventilated.

The arrangement of the station with the boiler room at one end is not advisable if conditions are such that the arrangement

shown in Fig. 339 is possible. When possible an arrangement with the pump room and boiler room beside each other will make a better plan for economy and for ease of superintending. In this case the plant is more compact and in a short time the chief engineer may have a complete view of the whole plant.

Fig. 352 shows a plant at Memphis, Tennessee, where the first vertical Worthington high duty pumps were installed. The plant was limited in ground area as a given capacity had to be placed in this space. The vertical high duty pump installed was of the compound duplex type, and on test in 1891, this pump gave 117,325,000 foot-pounds per 1,000 pounds of steam. The design was well thought out and as shown four 10,000,000 gallon pumps were placed in a pit 38 feet in diameter. The steam cylinders were 30 and 60 by 48 inches, while the water cylinders were 28 by 48 inches. The suction well is shown in the figure. The pit is founded on a heavy concrete base and the wall is built circular so that it need not be as thick as an ordinary retaining wall.

Fig. 353 shows the arrangement of the water works at Zurich, Switzerland, as described in the Engineering News, Vol. 32, page 34. This plant is operated by turbines, the vertical shafts of which drive the bevel wheels *E*, which are placed on short counter-shafts. These counter-shafts drive the main shaft *B* by means of the gears *D*, and the gears *C* drive the shafts of the pump which are blocked in at *AA*. The pumps are of the two-crank type, a crank being at each side of the pump with the pins quartering. These pumps are connected to four different services so that any pump can be used for any pressure. Of course this means that the pump power will change and this fact explains the reason for the long jack-shaft. A pair of pumps which would be operated by a pair of turbines properly at one pressure would not operate if it were desired to use a higher pressure. When such is the case, the additional power is obtained from other turbines. Although the use of the jack-shaft means a certain amount of friction, the flexibility of the station is increased so much by it that the installation is proper.

A small auxiliary boiler and engine may be used when the

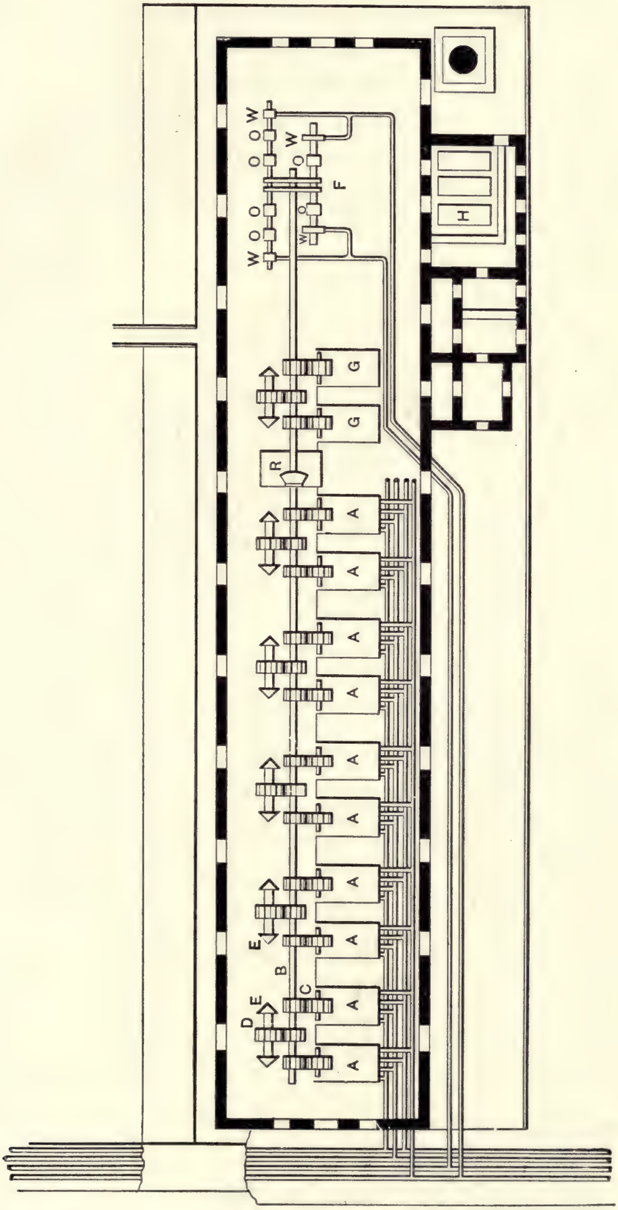


Fig. 353.—Zurich Station.

water wheels are out of commission. These are shown at *H* and *G*, and at *WW* are high pressure water wheels which may be

run from the high-pressure reservoir when needed to operate electric generators *OO* for lighting at night. The station also is connected to a rope transmission at *R*.

Fig. 354 shows the arrangement of air compressors for the Harris pump described in Chapter XIII. The steam cylinder

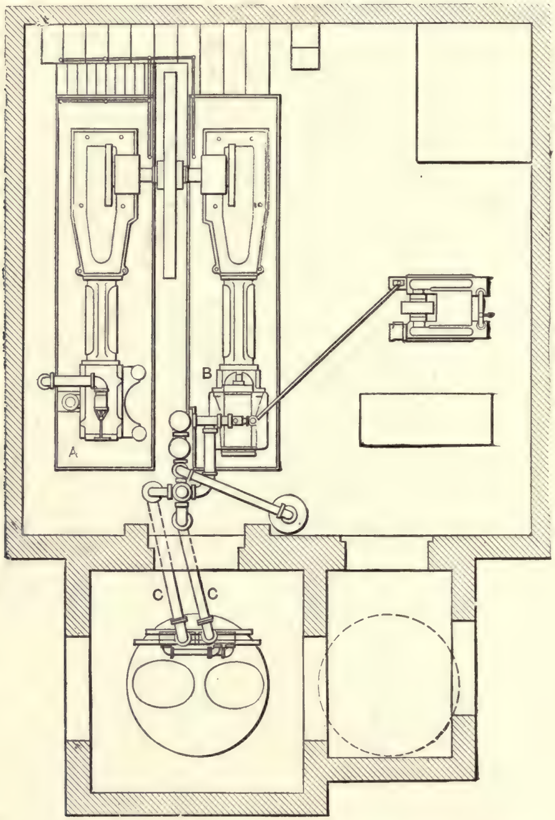


FIG. 354.—Harris Air Pump Station at High Bridge.

A receives steam from a boiler house not shown, while the air cylinder *B* discharges its air into one of the pipes *CC* and draws its supply from the other. The action of the pump is described later, but the plant is shown at this point so that the arrangement of machniery may be seen.

The station shown in Figs. 355 and 356 is located in Mil-

waukee, Wis., and is used for flushing the Milwaukee river. In the tower room are the cylinders of the steam engine, which is used to drive a screw pump which lifts 30,000 cubic feet of water (224,000 gals.) per minute against a head of $3\frac{1}{2}$ feet. The water is drawn through a tunnel from Lake Michigan and forced into Kinnickinnic River. The tunnel passes under the power house and turns after passing the screw of the pump.

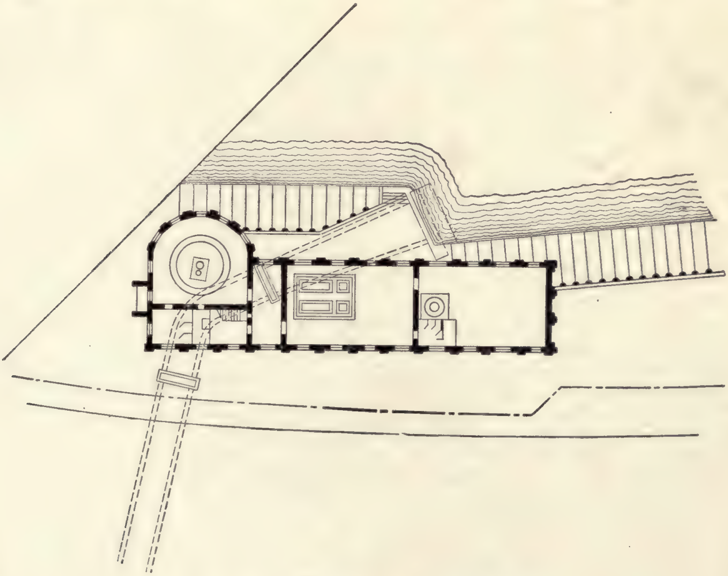


FIG. 355.—Kinnickinnic River Station.

The boiler house is equipped with two return tubular boilers furnished with superheaters. The engine is a tandem compound engine and the screw which was shown in Fig. 107 is $12\frac{1}{2}$ feet in diameter.

The boiler room and coal room are seen in the plan together with toilet room and locker room.

The Lardner's Point Pumping Station of the city of Philadelphia consists of three separate pumping stations, Figs. 357, 358 and 359. "The first is an old station formerly termed the "Frankford Pumping Station" and was used in the old system to pump water from the Delaware River to the Frankford dis-

tribution system. It is now termed No. 1 House, and the connection with the river has been closed and a new connection made to the filtered water conduit leading from the outlet shaft of the Torresdale conduit from the filter beds.

"No. 1 pumping station, or the old 'Frankford,' consists of one compound vertical Cramp pump of ten million gallons capacity, one Wetherill horizontal, ten million gallons capacity,

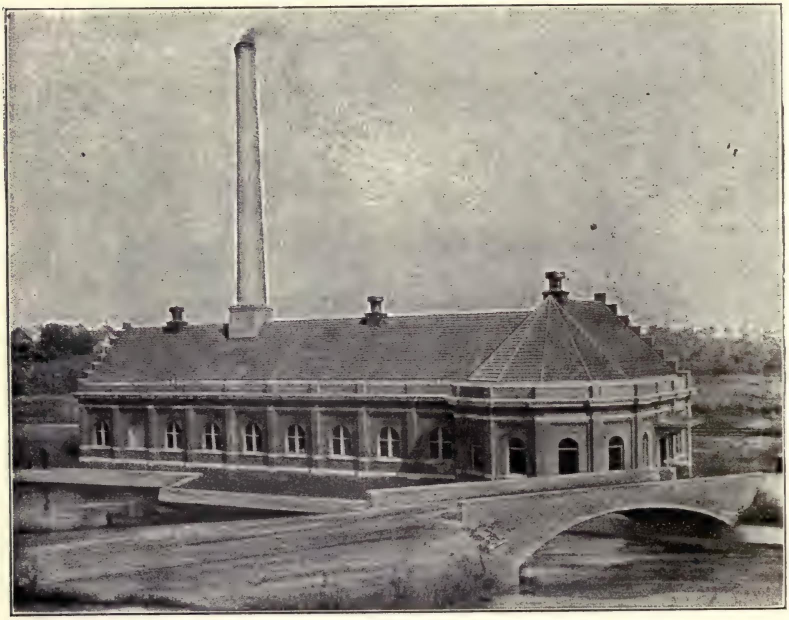


FIG. 356.—Kinnickinnic River Station.

one Southwark vertical, twenty million gallons capacity and one Southwark vertical horizontal, fifteen million gallons capacity. For this station there are twelve marine type boilers of 200 H.P. capacity each.

"Two entirely new stations were constructed and contain twelve (12) vertical triple expansion Holly pumping engines of twenty million gallons daily capacity each. The engine rooms are built separate from the boiler houses and are 171 feet long by 87 feet wide, constructed of gray standard size brick trimmed

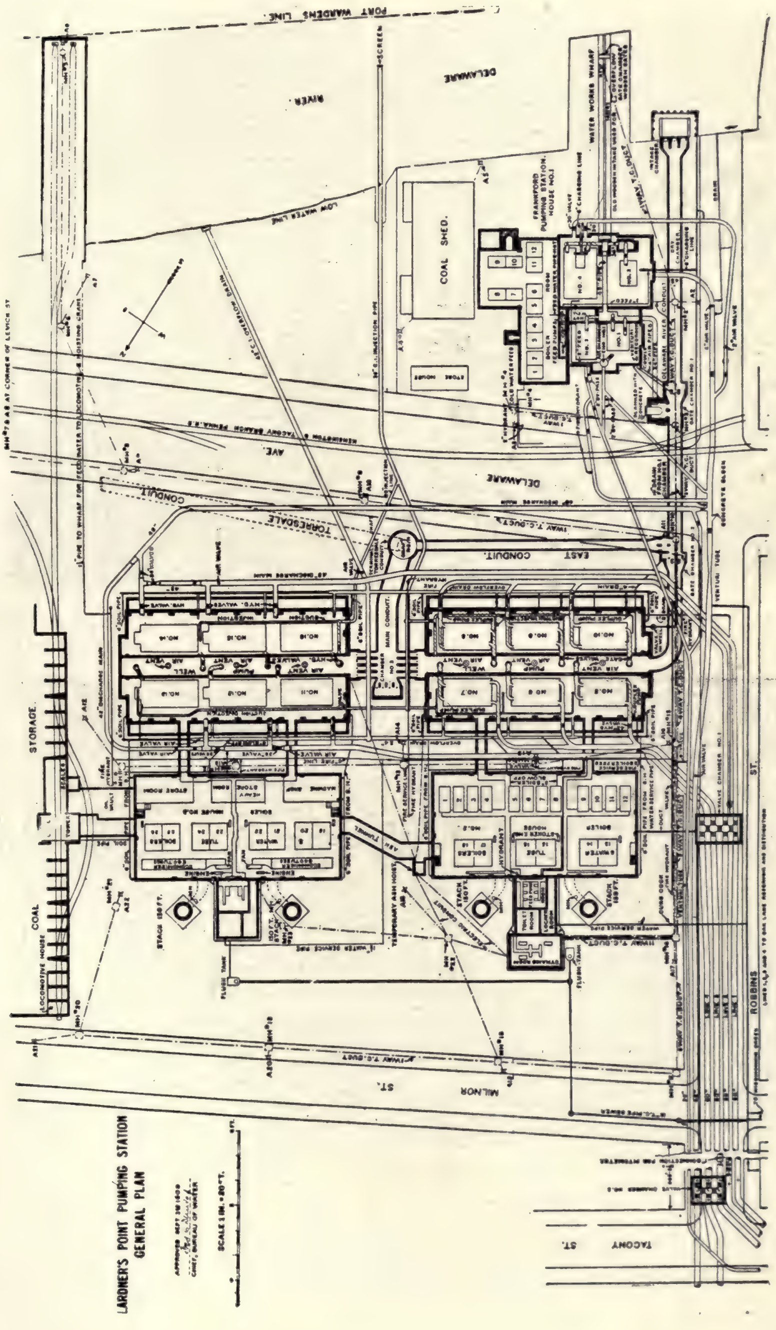


Fig. 357.—Lardner's Point Stations.

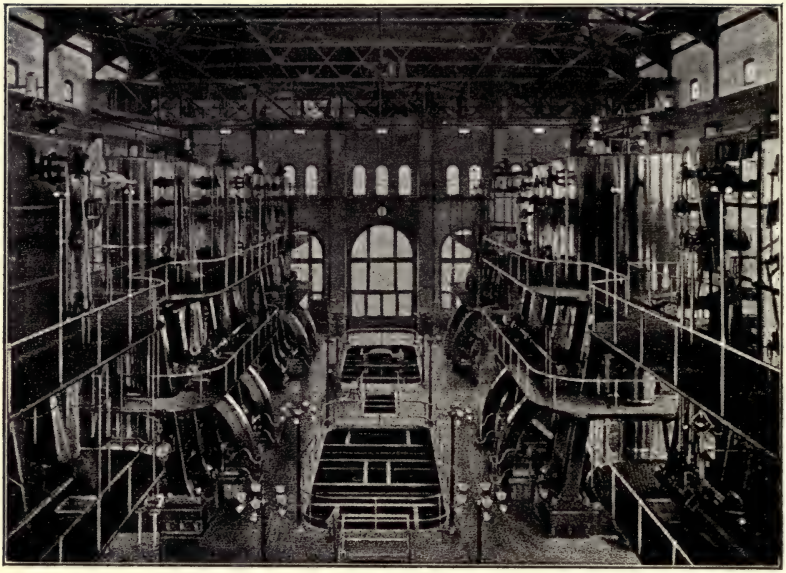


FIG. 358.—Lardner's Point Station.

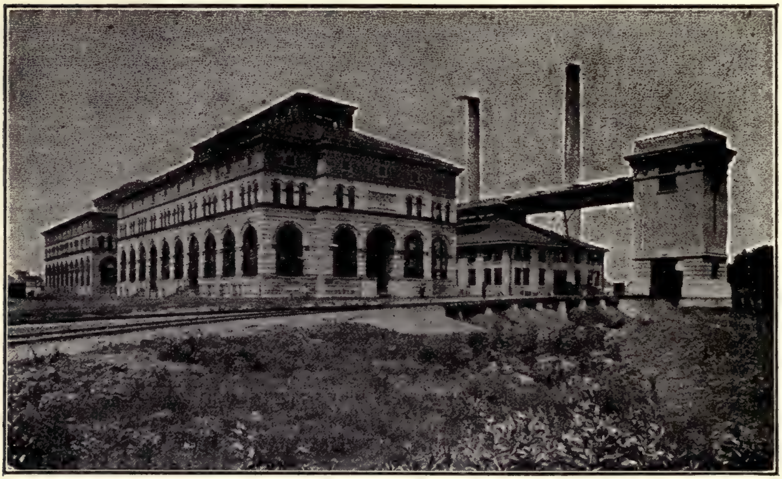


FIG. 359.—Lardner's Point.

with granite and terra cotta. All the roof coverings are of red tile. The water ends of the pumps are set in the basement under

the floor of the engine room, and the entire steam ends are all above the floor level. The pump well is located under the basement floor in the center of the engine houses, extending their full length. It is constructed of reinforced concrete, horseshoe shaped in section, 14 feet in width and height.

"Between engine houses Nos. 2 and 3 a gate chamber is located which controls the discharge from the larger connection to the outlet shaft of the Torresdale conduit. It is connected to the pump well of both houses, and gates have been installed for connecting the pump well of a future house to be located west of the present plant.

"The boiler houses of the new stations are of the same general architecture and contain the following:

House No. 2:

Six Edgemoor water tube boilers, 500 horse-power capacity each, equipped with Wetzel stokers.

Twelve Internal-fired tubular boilers, 200 horse-power capacity each.

House No. 3:

Eight Edgemoor boilers, 500 horse-power capacity each, equipped with Wetzel stokers and two Green economizers.

"In the annex of boiler house No. 2 are three 50 K.W. generators each, furnishing light for the entire station and current for the coal handling machinery, electric crane, etc.

"The smoke flues of the boiler houses are connected to four brick chimneys 150 feet high and 7 feet internal diameter—two for No. 2 and No. 3 houses of the Custodis and Heinicke type respectively.

"The coal is delivered to overhead pockets of 3,000 tons capacity in the boiler houses of Nos. 2 and 3 stations by means of a tower and belt conveyor with capacity for handling 50 tons per hour. Coal may be received either by rail or by boat, and the general design of this equipment is similar to the one at Torresdale.

"Data relating to engines and boilers in Nos. 2 and 3 houses, Lardner's Point pumping station:

ENGINES

Nominal capacity of each engine	20,000,000 gallons daily.
Number of revolutions per minute	20
Stroke	$5\frac{1}{2}$ ft.
Piston speed, feet per minute	220

	High.	Intermediate	Low.
Cylinder diameter	32″	60″	90″
Diameter piston rod	$7\frac{1}{2}$″	$7\frac{1}{2}$″	$7\frac{1}{2}$″

	First.	Second.
Receiver volume	205 cu.ft.	304 cu.ft.
Receiver heating surface	166 sq.ft.	304 sq.ft.

	Diameter.	Length.
Cross-head pins	12″	11″
Crank pins	12″	11″
Shaft bearings	$17\frac{1}{2}$″	32″
Shaft at center	$20\frac{3}{4}$″	——

Distance rods.—Four (4) each, 5 inches diameter.
Air pump.—One (1) 28-inch diameter, 66 inches stroke.
Feed pump.—One (1) $3\frac{1}{4}$-inch diameter, 66 inches stroke.
Feed water heater.—One (1) in exhaust, 308 sq.ft.
Fly wheels.—Two (2) 20 feet diameter and weighing 32 tons each (approximate).
Throttle valve.—8 inches diameter.
Exhaust pipe.—$24\frac{3}{8}$ inches diameter.
Suction pipe.—Main 42 inches diameter, branch 30 inches diameter.
Discharge pipe.—Main 42 inches diameter, branch 30 inches diameter.
Suction injection.—8 inches and 10 inches diameter.
Force injection.—3 inches and $3\frac{1}{2}$ inches diameter.
Overflow.—18 inches diameter.
Diameter of plungers.—33 inches.

	No. 2 House.	No. 3 House
Number of pump valves	960	864

BOILERS

Furnace Flue Tubular:

Number of boilers	12
Diameter of shell	108 ins.
Length of shell	20 ft.
Thickness of shell	$\frac{15}{16}$ in.
Number of fire boxes	2
Diameter in inches	41
Length of fire boxes	8 ft.
Number of tubes	195
Length of tubes	9 ft. 3 ins.
Diameter of tubes	$3\frac{1}{2}$ ins.
Length of grate	5 ft. $5\frac{3}{4}$ ins.
Area of grates	$40\frac{1}{2}$ sq.ft.

Water Tube Boilers:

Number of boilers	14
Stoker	Wetzel.
Number of tubes	252
Length of tubes	18 ft.
Diameter of tubes	4 ins.
Steam drums (two)	36 ins. diam.
Length of steam drum	21 ft.
Length of grate	$8\frac{1}{2}$ ft.
Area of grate	102 sq.ft.

Fig. 360 is an exterior view of the Queen Lane station of the Philadelphia water works and is shown to illustrate the appearance of a station.

Fig. 361 shows a Riedler water works pump used in the

FIG. 360.—Queen Lane Station.

water works of the city of Berlin, Germany. The large positively closed Riedler valves are seen and the tubes beneath the suction valves prevent the suction from breaking and make the upper part of the suction chamber an air chamber. The plunger is passed through a long sleeve and although different from the Worthington plunger and ring, it is the equivalent of it. The large air chambers on the discharge reduce the variation in water pressure.

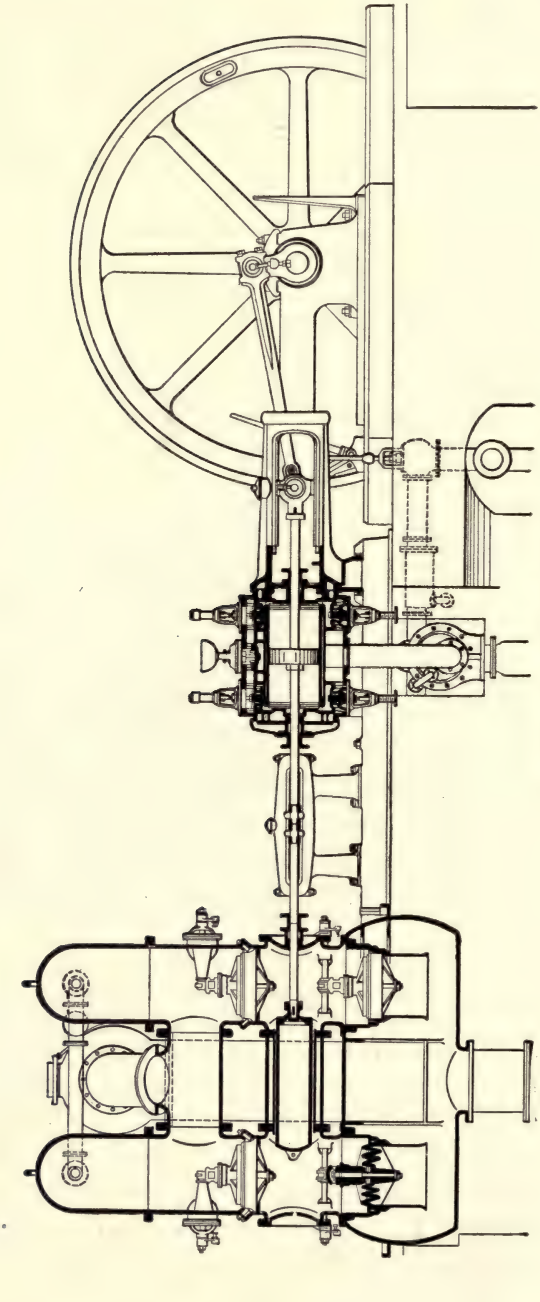


Fig. 361.—Riedler Pump.

The steam cylinders show the poppet valve form of cylinder so extensively used in Europe.

One of the Jersey City water works stations is equipped with this type of pump. The station is shown in Fig. 362.

The pumps are driven by turbines or steam cylinders placed in tandem with the water ends. There are three units installed in the station, two being high-pressure pumps and one a low-pressure pump. The suction is taken from the fore bay by pipes on each side of the unit and the two discharge air chambers are connected to the discharge main, which is carried to the end of the building.

Name.	Station.	Date.	Duty per 1,000 lbs. Dry Steam.
Savery	....	....	5,000,000
Smeaton Newcomen	....	....	12,000,000
Watt	....	....	20,000,000
Cornish engine	....	....	60,000,000
Simpson's	....	....	85,000,000
Holly Quadruplex	....	....	85,000,000
Gaskill	....	....	117,936,698
Worthington	....	....	120,000,000
Reynolds	....	....	118,327,041
Reynolds-Hannibal	....	....	154,048,700
Reynolds	North Point.	....	154,048,704
Reynolds	St. Louis.	....	179,454,250
Worthington (Direct)	Chicago.	Nov. 3-4, 1903	161,676,942
Worthington (Direct)	Chicago.	Aug. 27-28, 1902	174,734,801*
Allis-Chalmers	St. Louis.	....	181,000,000
Holly	Washington, D.C.	Nov. 2-3, 1905	162,644,000
Holly	Cleveland, Ohio.	Dec. 5, 1907	164,642,226
Holly	Brockton, Mass.	Oct. 14-15, 1909	170,000,000
Holly	Louisville, Ky.	May 1-2, 1909	195,020,000*
Holly	Albany, N. Y.	May 29-30, 1909	182,281,000
Snow	Mahanoy City, Pa.	Nov. 12, 1906	141,369,439
Allis-Chalmers	St. Louis.	Feb. 26, 1900	179,454,255
Allis-Chalmers	Boston, Mass.	May 1-2, 1900	178,497,000
Holly	Philadelphia, Pa.	Mar. 9, 1910	184,476,200
Worthington (Direct)	Fall River.	Sept. 23, 1909	136,500,000
Worthington (Direct)	Montreal.	Nov. 27, 1909	177,538,000*

*Superheat

The 12 inch steam pipe and the 24 inch exhaust pipe are carried in a trench on one side of the station and the exhaust from all the engines is cared for by one large jet condenser.

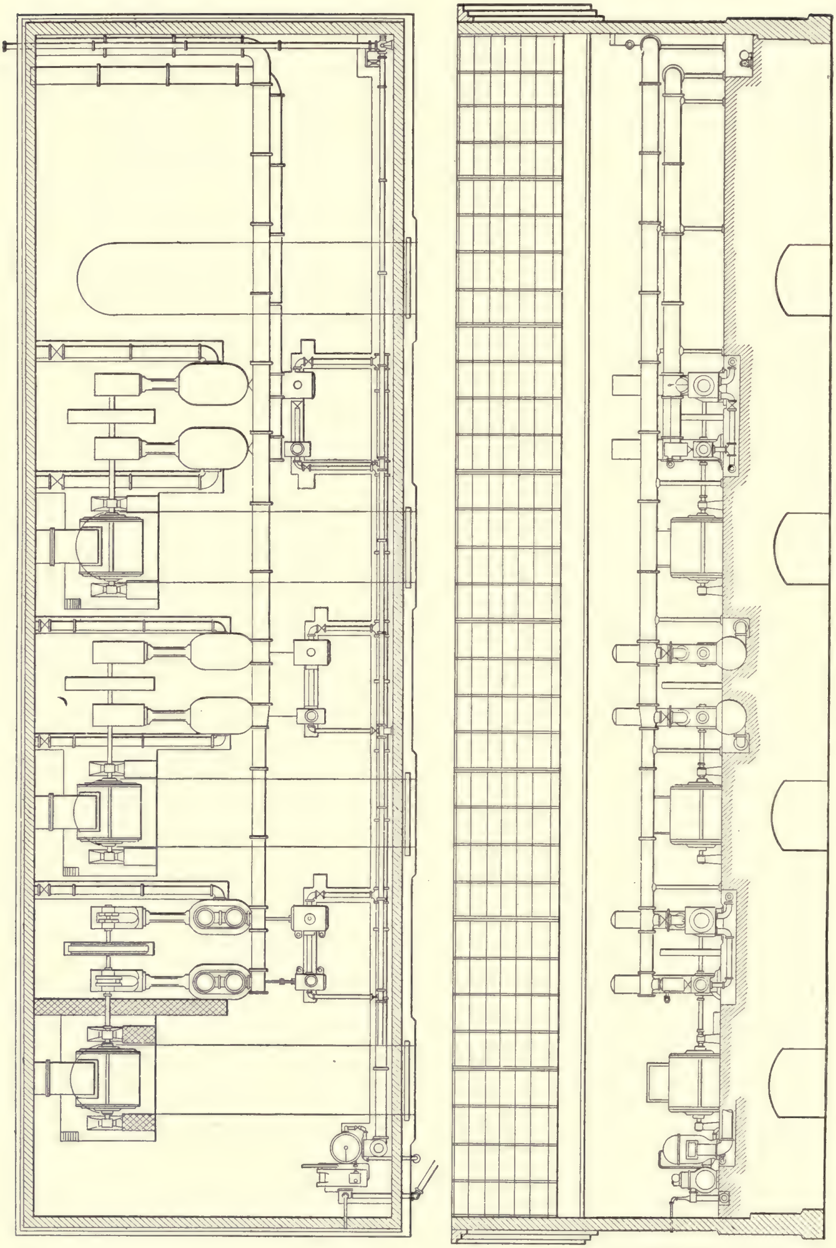


FIG. 362.—Jersey City Station.

In such a plant the engines are only used in case of low water, and at other times the pump rods are disconnected from the steam piston rod so that the piston does not act as a resistance to motion.

To give some idea of the value of the duty of modern pumping engines, see table on page 453.

CHAPTER XI

SPECIAL PUMPING MACHINERY

THERE are several special pumps used to handle liquids which should be discussed. These, although the same in principle as those which have been described, have particular features which make them noteworthy.

Condenser Pumps are of two kinds: Circulating pumps and air pumps. The circulating pump is used to force the condensing water through the condenser, while the air pump is used to remove the condensed steam and air from the steam side of a condenser, or, in the case of a dry air pump, the pump handles air and water vapor only, the condensed steam being removed from the condenser by a wet pump or by gravity as in a barometric condenser. The circulating pump in many cases lifts the water a short distance only, and for that reason it is built as a tank pump, while in other cases it may be used to overcome the friction of a long pipe line to and from the cooling tower as well as the lift to the top of the tower, in which case it must be heavier.

The circulating pump most often takes the form of a centrifugal pump, as it has to lift large quantities of water, and space is limited.

Fig. 363 shows a combined air and circulating pump of the Wheeler Condenser and Engineering Company. In this pump the two water cylinders are placed in tandem with the steam cylinder which takes the form of any of the simplex pumps.

In Fig. 363 the Knowles pump is shown. The circulating pump forms one end support for the condenser. The water is discharged through *A* into one set of tubes and then it returns through *B* and the upper set of tubes to *C*, where it discharges. The air pump forms the other support for the shell. It takes the air and water from the condenser and discharges it through *D*. The suction space *F* is connected to *G*.

To find the size of the water and air ends of the pump, suppose that W pounds of steam per hour at a pressure p are to be condensed. If r is heat of vaporization of the steam, x its quality, $t_c°$ the temperature of the condensed steam, and q the heat of the liquid, and if G pounds of water entering at

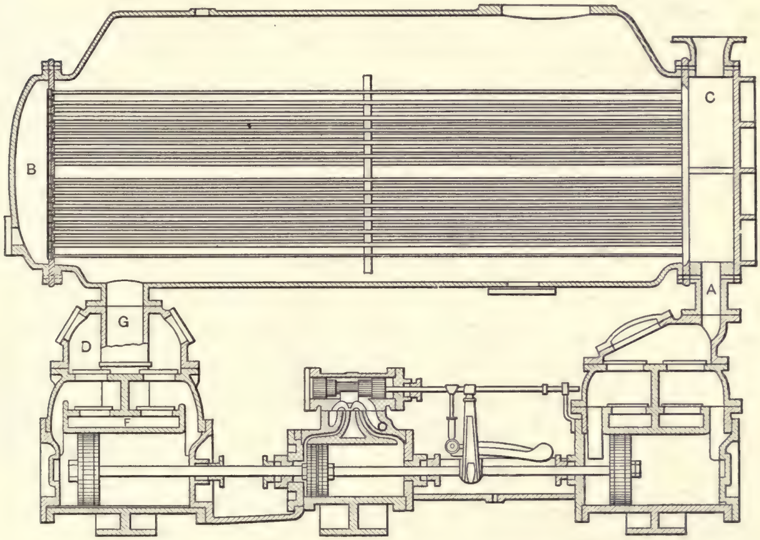


FIG. 363.—Combined Air and Circulating Pump.

$t_i°$F and leaving at $t_0°$ F are to be used, G is given by the equation:

$$G = \frac{W(q + xr - q_{tc})}{q_{t_0} - q_{ti}}.$$

If the number of revolutions or double strokes N are assumed, the displacement of the water end will be

$$D = \frac{G}{120N \times 62.5}.$$

The air end of the pump is made in many cases of empirical design. Some authors give ratios of volume displaced by the pump per minute to the volume of the condensed steam or to the volume of the low-pressure cylinder of the engine which is discharging into the condenser. Several of these are mentioned.

RATIO OF AIR CYLINDER DISPLACEMENT TO LOW-PRESSURE CYLINDER.

Single-acting vertical pump surface condenser	1 : 9
Single-acting vertical pump jet condenser	1 : 13
Double-acting horizontal pump surface condenser	1 : 15
Double-acting horizontal pump jet condenser	1 : 12
Double-acting horizontal pump-compound engine surface condenser	1 : 26
Single-acting horizontal pump-compound engine surface condenser	1 : 16

RATIO OF AIR CYLINDER DISPLACEMENT TO VOLUME OF CONDENSED STEAM.

Surface condenser	1 : 20
Jet condenser	1 : 40

This may be a satisfactory way, but it is better to estimate the volume from the air probably present. Water usually contains air to about one-fifteenth of its volume. This amount of air is at atmospheric pressure p_a and it must be cared for by the air pump at a reduced pressure. In addition to this there are small leaks in the pipe line which allow more air to enter. A small hole will destroy the proper acting of the air pump. The author at one time desired to run a 2000 H.P. condensing engine with no vacuum and still use the condenser to operate so as to measure the steam consumption, and he found that a 2-inch plug removed from a 30 inch exhaust main was sufficient to destroy the vacuum completely with the air pumps running at full speed. To find the volume of air per minute the following formula will be used, allowing 100 per cent for leakage.

$$V=\left(2\times\tfrac{1}{15}\times\frac{W}{62.5}\right)\left(\tfrac{1}{60}\right)\frac{14.7T_c}{(p-p_s)T_a}.$$

p = absolute pressure in the condenser, lbs. per sq.in.

p_s = vapor tension or absolute steam pressure corresponding to T_c

T_c = absolute temperature in condenser

T_a = absolute temperature atmosphere.

This equation shows the importance of making p_s as different from p as possible. The terms p and p_s do not differ much, and by taking the mixture of air and vapor on its way to the air pump, through as cold a passage as possible, the term p_s is made smaller and the denominator is increased, making V small. This is the reason for the great advantage in counter current for condensers, and even in the condenser, shown in Fig. 363, the coldest water should enter directly over the air pump inlet so as to cool the mixture going to the pump.

From the volume thus computed the displacement of the air pump is given by:

$$D_{ap}=\frac{V}{120N}.$$

Knowing the displacements of these pumps a stroke may be assumed, and from it the area determined.

$$A_p=\frac{D_p}{L},$$

$$A_{ap}=\frac{D_{ap}}{L}.$$

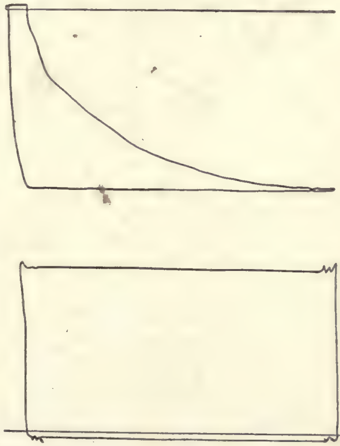


FIG. 364.—Cards from Air Pump and Circulating Pump.

The cards from the water end are shown in the lower part of Fig. 364, while those for the air end are shown above. The combination of these or the addition of them when reduced to the proper scale, on account of the difference in piston area, will give the total work, and from this the size of the steam cylinder is given, if the M.E.P. be found for a given boiler pressure. Allowing 33 per cent for friction, which is made large to give a certain driving power, the following results:

$$A_{sc}=\frac{(\text{M.E.P.})_{ac}A_{ac}+(\text{M.E.P.})_pA_p}{(1.00-0.33)(\text{M.E.P.})_s}.$$

Separate air pumps are often used. Fig. 365 shows a steam driven pump used in the U. S. Navy. This air pump is

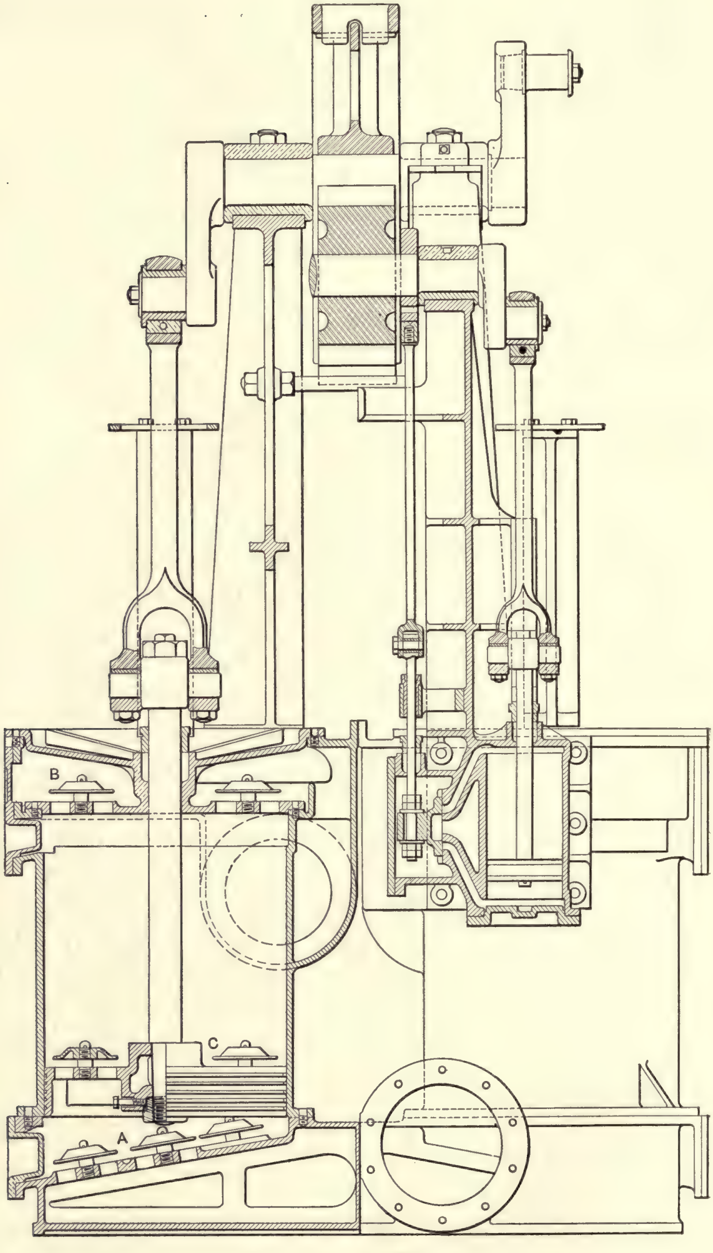


FIG. 365.—Air Pump.

made with two air cylinders driven through gears from a steam cylinder placed on one side of a pump barrel. The pump is of the bucket type with foot valves *AA* and head valves at *B*. These with the valves in the bucket at *C* are all spring-controlled metal valves. The foot valves are placed on an inclined partition for the purpose of making it easier to discharge the air when the piston rises and forms a vacuum. The lip around the discharge valve makes a dam and covers the valve with water. This makes them air tight. The other valves are in that condition, since all of the water on the bucket or that over the foot valves can not be driven out, as the valves limit the motion of the bucket. On the down stroke of the bucket the pressure in the space above it soon falls to a low vacuum because it had been completely filled with water; this, then, causes the valves to open and take air from the lower portion of the cylinder. The air in the water also separates and rises to the top of the cylinder. Finally the bucket reaches the water below, and this is driven through the valve openings which are uncovered. It is seen that the air leaves first in this case; the water is struck by the bucket surface and will cause considerable shock if the pump is running too rapidly.

To do away with shock and to decrease valve resistance, the Edwards Air Pump, Fig. 366, was introduced. In this air pump water and air enter the space *A* at the bottom of the pump which is made conical in form. The piston *B* which is driven from the steam piston in *I* by two rods *CC* extending over the shaft and crank, is provided with a conical bottom. As this piston descends there is a vacuum produced, so that when the top of the piston uncovers the openings *EE*, air enters from the space *A* around the cylinder barrel, and as the conical bottom enters the water in the bottom of *A*, this is forced around the curved passage and discharged into the openings at *E*. This continues even after the piston starts up, as the momentum of the water continues its motion. This discharge of water into the openings as the piston is moving upward acts as a valve to keep the air from coming out as the piston ascends. In a short

time, however, the piston covers the ports or openings E and then the air and water are compressed until the pressure is sufficient to overcome the atmospheric pressure on the head valves at H, which are drowned by the use of a lip around the valve deck. The piston rods CC are carried through long-sleeve

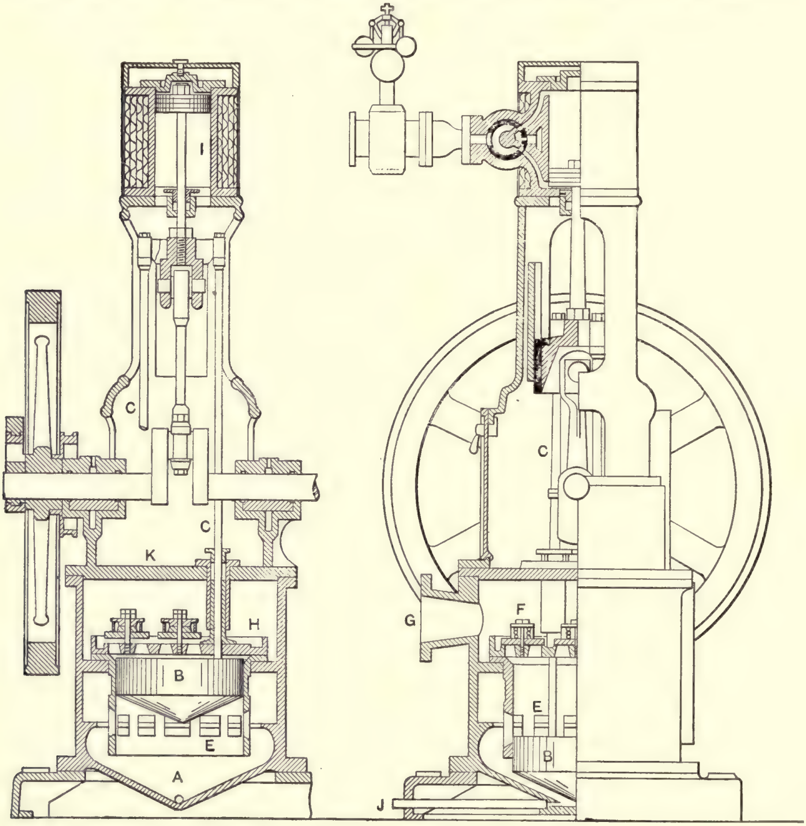


FIG. 366.—Edwards Air Pump.

stuffing boxes so arranged that the point H, at which leakage could occur, is water sealed, leaving only one stuffing box at the plate K to care for. This is a simple matter.

The Mullen Valveless Air Pump, Fig. 367, is somewhat similar to the Edwards Pump. In this case, the deep piston C, provided with a number of packing grooves, uncovers ports

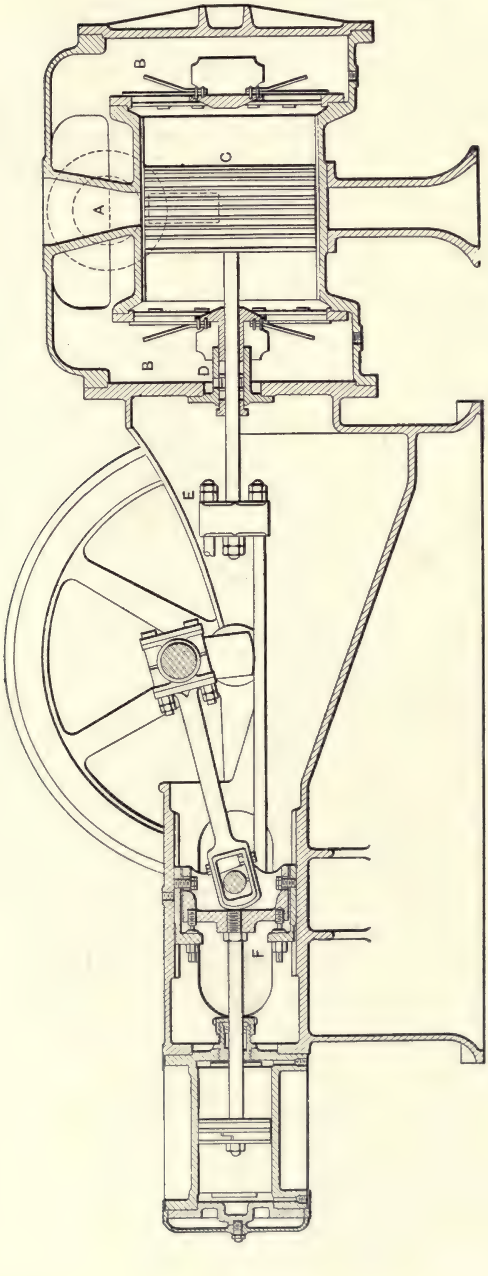


FIG. 367.—Mullen Air Pump.

at the center of the cylinder. The vacuum formed by the motion of the piston from the end draws in the air and water from the space *A* around the cylinder barrel and the return of the piston cuts these ports off and compresses the air against the spring valves at the end, which open after the atmospheric pressure is reached. The spaces *BB* unite and lead to the hot well.

The piston of this pump is of such a length that when the ports of *A* are completely uncovered, the piston just reaches the end of its travel, leaving a small amount of clearance which is filled with water, so that as soon as the piston leaves the end of its stroke the pressure falls. This action is the same in all wet air pumps. The piston rod of the Mullen Air Pump is attached to a cross head *E* which is connected to the cross head *F* of the steam end by two rods, so that the shaft and crank may be cleared. The stuffing box at *D* is water sealed to cut down the leakage of air. Air leaks, even the smallest, are to be avoided on account of the low pressure in the pump.

Dry air pumps have become quite common. They were first introduced for barometric condensers and afterwards for use with surface condensers. As a type of this class, the Alberger Air Pump, Fig. 368, is shown. In it the piston *A* travels from end to end and is so finished on the ends that its clearance is small. A rotary valve below the spring discharge valve *D* directs the air to the suction chamber *C* or to the discharge. This valve *B* is positively driven. As shown in the left hand figure, the air on the left is being drawn from *C*. The air on the left is not discharged until it has reached a pressure slightly above the atmosphere, when it can raise the spring valve *D* and escape. From this point the air is driven out as the piston continues to move to the left. When the end of the stroke is reached the pressure on one end is atmospheric and that on the other is that of the maximum vacuum carried on the condenser. The piston will draw no more on this stroke and the vacuum of the condenser would not be effected if air was allowed to enter this side of the piston. The other side of the piston has air at atmospheric pressure, filling the clearance volume,

including the passage E. If this air remained in the clearance volume, it would cut down the amount of air drawn in, as none would enter until it had reached the pressure of the condenser, if the suction had a valve; or, it would change the vacuum if allowed to discharge back into the condenser. To obviate this, the Alberger Company introduced a small cross connection passage in their rotating valve which connects the two ends of the cylinder when the piston is just at its dead point. This arrangement is shown on the right of Fig. 368, where the piston

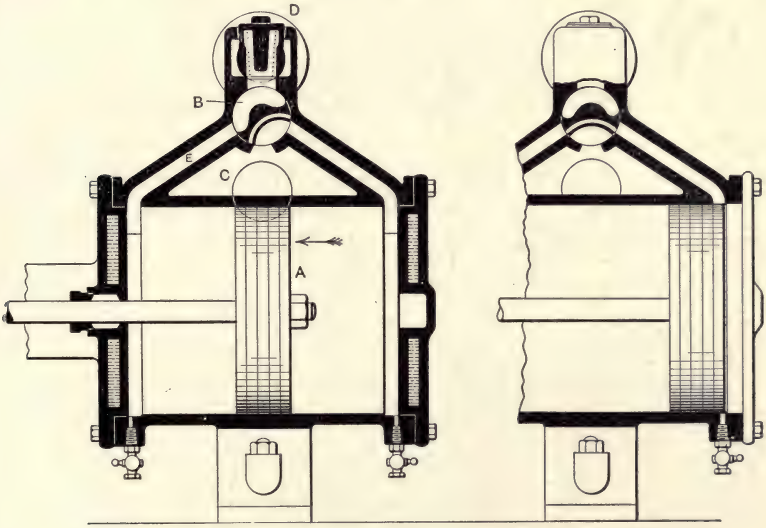


FIG. 368.—Alberger Air Pump.

is at the right hand end of its stroke. In this position the space to the right of the piston, as far as the valve B, is filled with air at atmospheric pressure, while the space on the left of the piston is at condenser pressure and has reached the point where it will not receive any more air from the condenser. If now the valve cuts off both of these ends from the air and condenser but connects them through the small passage in the valve, the pressure on the right will be reduced very materially, as the volume of the left hand end of the cylinder is so large compared with the volume of the clearance. The pressure in

the clearance is then reduced to vacuum pressure when the valve opens the right side to the condenser, and the left hand end of low vacuum has received a little more air to be discharged.

This valve passage therefore makes the volumetric efficiency of the pump greater, although the power required to drive the pump for a given quantity is not changed. Fig. 369 shows the card from such a pump compared with one without this arrangement. The solid curve shows how the clearance effect is reduced and how the quantity of air handled is made greater. The curve also shows that the power required in compressors of the same displacement is increased. The volume of air handled per stroke is increased from *AB* to *CB*.

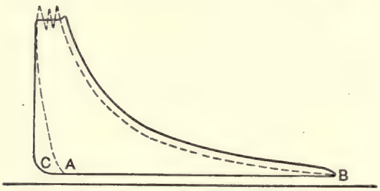


FIG. 369.—Alberger Air Pump Card.

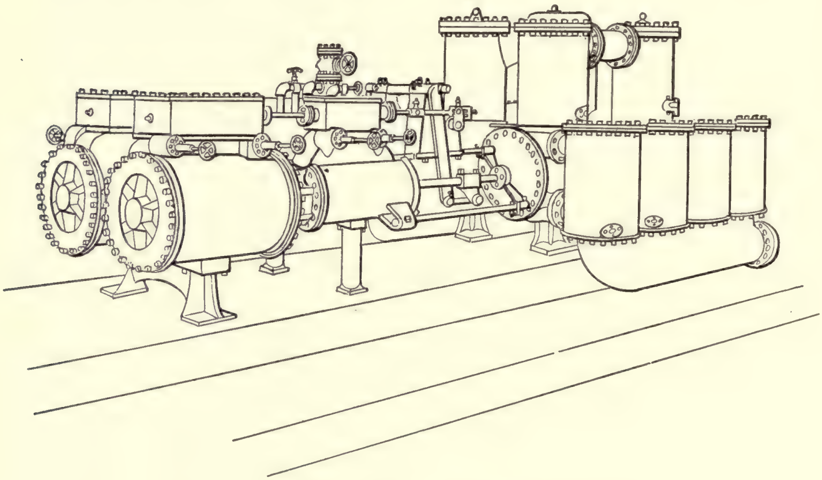


FIG. 370.—Sewage Pump.

The small curve at *C* is due to the pressure not falling to the vacuum of the condenser when the cross connection is made, while the rise in pressure at *B*, giving a starting point of com-

pression above the vacuum pressure, is due to this cross connection.

Fig. 370 illustrates a pump built by the Laidlaw, Dunn, Gordon Co. for the pumping of sewage. The valves are made large on account of the solid matter in suspension, and they are placed in large valve boxes on the sides of the pump. The pump is otherwise similar to any duplex compound pump.

The Underwriters' Pump, or better, the "National Standard

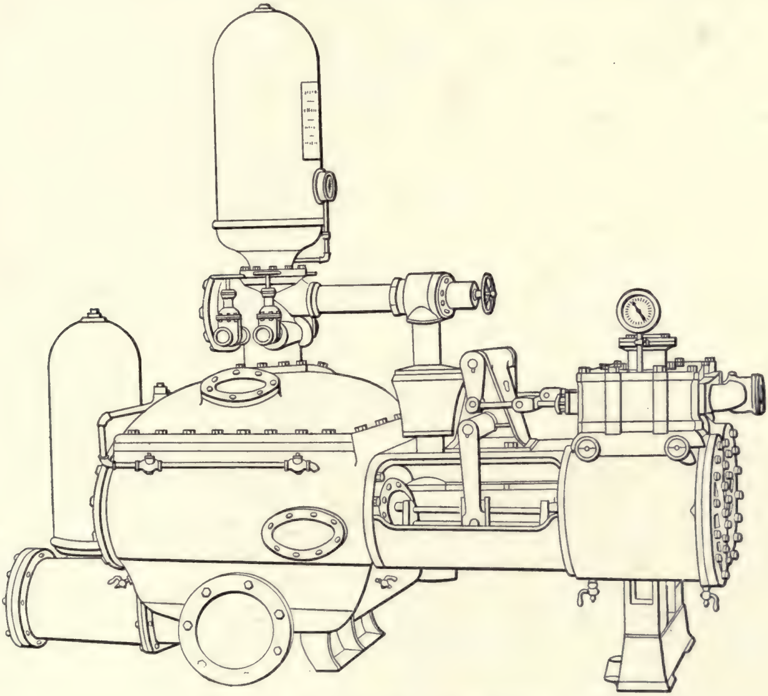


Fig. 371.—Underwriters' Pump.

Fire Pump," to which attention has been called in Chapter II, is illustrated again in Fig. 371. This figure illustrates the form used in accordance with the specifications of the National Board of Fire Underwriters, and to give the important parts of these specifications the following has been taken from their pamphlet.

"SPECIFICATIONS OF THE NATIONAL BOARD OF FIRE UNDERWRITERS FOR THE MANUFACTURE OF STEAM FIRE PUMPS," EDITION OF 1908.

UNIFORM REQUIREMENTS.

The following specifications for the manufacture of Steam Fire Pumps, developed from those originally drawn by Mr. John R. Freeman, are now used throughout the whole country, having been agreed upon in joint conference by representatives of the different organizations interested in this class of work. They will be known as "The National Standard," and have been up to this time adopted by the following associations:

Associated Factory Mutual Fire Insurance Companies.
National Board of Fire Underwriters.
National Fire Protection Association.

NATIONAL STANDARD SPECIFICATIONS FOR THE MANUTURE OF STEAM FIRE PUMPS

1. *Workmanship.* *a.* The general character and accuracy of foundry and machine work must throughout equal that of the best steam-engine practice of the times, as illustrated in commercial engines of similar horsepower.

2. *Duplex Only.* *a.* Only "Standard Duplex pumps" are acceptable.

So-called "Duplex" pumps, consisting of a pair of pumps with "steam-thrown valves" actuated by supplemental pistons, are not acceptable.

Further, the direct-acting duplex has the great advantage over a fly-wheel pump of not suffering breakage if water gets into steam cylinder.

3. *Sizes of Pumps.* *a.* Only the four different sizes given in the table on page 469 will be recognized for "National Standard" pumps.

b. The tabular sizes of steam and water cylinders and length of stroke have given general satisfaction and will now be considered as standard.

A steam piston relatively larger than necessary is a source of weakness. It takes more volume of steam, and gives more power with which to burst something if the throttle is opened wide suddenly during excitement.

It has been common to make all fire pumps with water plunger of only one-fourth the area of steam piston, with the idea that the pump could thereby be more readily run at night, when steam was low. The capacity in gallons is thus reduced 25 per cent as compared with a 3 to 1 plunger on the same steam cylinders.

Often, especially with large pumps, "4 to 1" construction is a mistake, and gives no additional security, although the pump might start and give a few puffs with 30 lbs. of steam on banked fires; because, if any pump of whatever cylinder ratio draws 50 or 100 horse-power of steam from boilers with dead fires, it can run effectively only a very short time (ordinarily, perhaps, 3 to 5 minutes), unless fires are first aroused to make fresh steam to replace that withdrawn.

Steam pressures stated above must be maintained *at the pump,*

NATIONAL STANDARD PUMP SIZES

Pump Sizes.			Ratio of Piston Areas.	Capacity at 100 lbs., at Pump.			Boiler Power Required.		Full Speed.	
Steam.	Water.	Stroke.	About	Number of $1\frac{1}{8}$-in. Streams.	Nominal gals. per minute.	Actual gals. per min. as per Art. 4.	Horse-power.	Steam Pressure at Pump, lbs.	Revolutions per minute.	Piston Travel, feet per minute.
14	x 7	x12	4 to 1	Two	500	483	100	40	70	140
14	x $7\frac{1}{4}$	x12				520				
16	x9	x12	3 to 1	Three	750	806	115	45	70	140
18	x20	x12	3 to 1	Four	1000	999	150	45	70	140
$18\frac{1}{2}$	x $10\frac{1}{4}$	x12				1050				
20	x12	x16	$2\frac{3}{4}$ to 1	Six	1500	1655	200	50	60	160

to give full speed and 100 lbs. water pressure. Pressure at boilers must be a little more, to allow for loss of steam pressure between boiler and pump. Pumps in poor order, or too tightly packed, will require more steam.

c. Two hundred and fifty gallons per minute is the standard allowance for a good $1\frac{1}{8}$-inch (smooth nozzle) fire stream.

A so-called "Ring Nozzle" discharges only three-fourths as much water as a smooth nozzle of the same bore, and is not recommended.

From fifteen to twenty automatic sprinklers may be reckoned as discharging about the same quantity as a $1\frac{1}{8}$-inch hose stream under the ordinary practical conditions as to pipes supplying sprinkler and hose systems respectively.

4. *Capacity.* *a.* Plunger diameter alone will not tell how many gallons per minute a pump can deliver, and it is not reasonable to continue the old time notion of estimating capacity on the basis of 100 feet per minute piston travel.

b. The capacity of a pump depends on the speed at which it can be run, and the speed depends largely on the arrangement of valves and passageways for water and steam.

c. It is all right to run fire-pumps at the highest speed that is possible without causing violent jar, or hammering within the cylinders. Considerations of wear do not affect the brief periods of fire service or test, hence these speeds are greater than allowable for constant daily duty.

d. Careful experiments on a large number of pumps of various makes at full speed, show that in a new pump with clean valves, and an air-tight suction pipe, and less than 15 feet lift, the actual delivery is only from $1\frac{1}{2}$ to 5 per cent less than plunger displacement. This slip will increase with

wear, and for a good average pump in practical use, probably 10 per cent is a fair allowance to cover slip, valve leakage, slight short-stroke, etc.

e. Largely from tests, but partly from "average judgment," and recognizing that a long stroke pump can run at a higher rate of piston travel in lineal feet per minute than a short stroke pump, and that a small pump can make more strokes per minute than a very large one, the speeds given in the preceding table have been adopted as standards in fire service for direct-acting (non-fly-wheel) steam pumps, which have the large steam and water passages herein specified.

f. Rated capacity is to be based on the speed in the preceding table, correcting the plunger displacement for one-half the rod area and deducting 10 per cent for slip, short-stroke, etc.

5. *Capacity Plate.* *a.* Every steam fire-pump must bear a conspicuous statement of its capacity securely attached to the inboard side of air chamber, thus:

NATIONAL STANDARD FIRE PUMP

16 × 9 × 12

CAPACITY

750 GALLONS PER MINUTE, OR THREE GOOD $1\frac{1}{8}$-IN. SMOOTH NOZZLE FIRE STREAMS

FULL SPEED

70 REVOLUTIONS PER MINUTE

Never let steam get below 50 *pounds, nights, Sundays, or at any other time*

b. This plate must have an area of not less than one square foot, and must be made of an alloy at least 85 per cent aluminum and the remainder zinc. The letters must be at least one-half inch in height, plain and distinct, with their surfaces raised on a black background and buffed off to a dead smooth finish.

c. A smaller plate of composition must be attached to steam chest bearing the size of pump, the shop number, and the name of shop in which the pump was built.

6. *Strength of Parts.* *a.* The maker must warrant each pump built under these specifications to be, at time of delivery, in all its parts strong enough to admit of closing all valves on water outlet pipes while steam valve is wide open and steam pressure eighty pounds, and agree to so test it before shipment from his works.

b. The pump must be warranted so designed and with such arrangement of thickness of metal that it shall be safe to instantly turn a full head of steam on to a cold pump without cracking or breaking the same by unequal expansion.

7. *Shop Inspection.* A systematic shop inspection must be given to each pump to ensure complete workmanship, and to prevent the use of defective parts, improper materials, or the careless leaving of foreign matter in any part of the cylinders or chests.

THE STEAM END.

8. *Steam Cylinders.* *a.* These must be of hard close iron with metal so distributed as to insure sound castings and freedom from shrink cracks. The following are the minimum thicknesses acceptable:

14″ Diam.	$\frac{7}{8}$″ thick.	18″ Diam.	1″ thick.
16″ "	$\frac{15}{16}$″ "	20″ "	$1\frac{1}{8}$″ "

b. The inside face of the steam cylinder heads and the two faces of the piston must be smooth surfaces, fair and true, so that if the piston should hit the heads it will strike uniformly all around, thus reducing to a minimum the chances of cramping the piston rod or injuring the pump.

c. All flanged joints for steam must be fair and true and must be steam tight under 80 pounds pressure if only a packing of oiled paper 1-100 inch thick covered with graphite were used. Jenkins, "Rainbow" or equivalent packing of not exceeding $\frac{1}{32}$ inch original thickness is acceptable. Oiled paper is not acceptable as a final packing, as it burns out.

For size of steam and exhaust pipes, standard flanges and bolting, see Art. 39.

d. Heads at both ends of cylinder must be beveled off very slightly over a ring about one inch wide, or equivalent means provided to give steam a quick push at piston, should it stand at contact stroke.

The specifications originally required machine facing for all these surfaces. The art of machine molding from metal patterns with draw plates, etc., has, however, attained such excellence in certain shops, that in regular practice "foundry faced" cylinder heads and piston faces can be made true and fair, and steam joints can be made tight under 80 lbs. pressure with a packing of oiled paper only, 1-100-inch thick.

11. *Steam Ports.* *a.* The area of each exhaust steam passage, at its smallest section, must not be less than 4 per cent of the area of the piston from which it leads.

b. Each admission port must be not less than $2\frac{1}{2}$ per cent of area of its piston and to avoid wasteful excess of clearance, these passages should not be bored out larger in interior of casting than at ends or passage.

c. The edges of the steam-valve ports must be accurately milled, or chipped and exactly filed to templets, true to line, and the valve seat must be accurately fitted to a plane surface, all in a most thorough and workmanlike manner and equal to high-grade steam-engine work.

d. To guard against a piston ring catching in the large exhaust ports, these ports must have a center rib with cylinder at cylinder wall. See also Art. 13 *d.*

12. *Steam-clearance Space.* *a.* Clearance (including nut-recess, counter-bore, and valve passages) must not exceed 5 per cent for contact stroke or about 8 per cent for nominal stroke (i.e., contact stroke should overrun nominal stroke at each end about one-half inch).

b. The clearance space between face of piston and cylinder head must be reduced to smallest possible amount, and these contacting surfaces be

flat, without projections or recesses other than the piston rod nut and its recess.

Some makers, with the idea that a fire pump need not be economical, have not taken pains to keep these waste spaces small.

Securing small clearance costs almost nothing but care in design, and is often of value, since at many factories boiler capacity is scant for the large quantity of steam taken by a fire pump of proper size.

13. *Steam Pistons.* *a.* May be either built up or solid, as maker thinks best.

It is believed that "solid" (cored) pistons with rings "sprung in," are for fire pumps much preferable to built-up pistons, since follower bolts *do* sometimes get loose.

b. The thickness of piston should be about one-fourth of its diameter. If solid, walls should be not less than ½ inch thick, and special care should be given to shop inspection to insure uniformity of thickness.

This will demand, for the four sizes of pumps, pistons as follows:

500-gal.	750-gal.	1000-gal.	1500-gal.
Diameter 14 in.	Diameter 16 in.	Diameter 18 in.	Diameter 20 in.
Thickness 3½ in.	Thickness 4 in.	Thickness 4½ in.	Thickness 5 in.

Manufacturers desiring to use existing patterns approximating these thicknesses may be allowed to do so after due consideration of working drawings.

c. If built-up pistons are used, involving follower bolts, such bolts must be of best machinery steel, with screw thread cut for about twice the diameter of the bolt and fitting tightly its whole length.

d. The width of each piston ring must exceed the length of the large exhaust port by at least ¼ inch.

This is to avoid the possibility of piston ring catching in the port.

See also Art. 11 *d.*

14. *Steam Slide-valves.* *a.* Slide valves must be machine fitted on all four of the outer edges, the exhaust port edges, and the surfaces in contact with rod connections.

b. The slide valve itself must have its steam and exhaust edges fitted up "line and line" with their respective steam and exhaust ports.

The adding of lap to these edges in lieu of lost motion is not acceptable further than a possible $\frac{1}{32}$ of an inch to cover inaccuracies of edges.

c. The valves must be guided laterally by guide strips cast in steam chest, and these strips must be machine fitted. The lateral play at these surfaces should not exceed $\frac{1}{16}$ inch. The height of these guide strips should not be less than ½ inch, measuring from valve seat.

The construction must be such as to absolutely preclude the possibility of the valve riding up on top of this guide strip.

d. The valves must be guided vertically by the valve-rod itself, the inside end of which must be kept in alignment by the usual form of tail-rod guide.

The vertical play at these parts should not exceed ⅛ of an inch.

e. The surface of valves must be machine faced and accurately fitted to a plane surface, and be steam tight when in contact with the seat of steam valve.

15. *Steam Slide-valve Adjustment.* *a.* The lost motion at the valves and

the settling of them must be determined by a solid hub on the rod, finished in the pump shop to standard dimensions, so that no adjustment is possible after the pump is once set up.

It is recognized that the practice of making adjustable valve tappets located outside of the steam chest is a good thing in a large pump in constant service and operated by a skilled engineer, but for the infrequently used ordinary fire pump. the utmost simplicity is desirable, and it is best not to tempt the ordinary man to readjust the valve gear.

The common form of lost motion adjustment consisting of nut and check nut at each end of the slide valve is not acceptable, as these nuts are liable to become loose and may be incorrectly reset by incompetent persons. A long rectangular nut in the center of the valve is also not acceptable, as it can be moved out of adjustment. A solid hub made as a part of the rod is required, as it absolutely avoids the possibility of the hub becoming loose, an accident possible with a separate hub attached to the rod.

The amount of lost motion should generally be such that admission takes place at about ⅝ of the stroke of the piston. i.e., for 12-inch stroke R.H. valve will be about to open when L.H. piston has moved 7½ inches to 8 inches from the beginning of stroke. When piston is at end of stroke the ports should be full open.

16. *Rock Shafts, Cranks, Links, etc.* *a.* Rock shafts must be either forged iron, forged steel, or cold rolled steel. Cast iron is not acceptable.

The following are the minimum diameters acceptable:

500 gallon pump	1½ in.
750 gallon pump	1¾ in.
1000 gallon pump	2 in.
1500 gallon pump	2 to 2¼ in.

b. The rock-shaft bearings must be bushed with bronze and the bushings pinned firmly in place. The length of each of these non-corrosive bearings must be not less than 4 inches.

c. Rock-shaft cranks, valve-rod heads, valve-rod links, and piston-rod spools or cross heads may be wrought iron or steel forgings, or steel castings. If of a heavy, strong pattern, these parts, with the exception of valve-rod links, may be of semi-steel or cast iron.

d. The sectional area of all connections between rock-shaft cranks and valve rod must be such as to give a tensile or compressive strength substantially equal to that of the valve rod.

17. *Valve-motion Levers.* *a.* The valve-motion levers must be steel, wrought iron, or steel castings. Cast iron is not acceptable. Steel castings, if used, must be deeply stamped with the name of the makers, with letters one-eighth inch high, near the upper end of each lever, where it can easily be seen,—thus "*--Steel Castings.*"

Cast-iron arms, if bulky enough to be safe against external blows, are awkward in shape. The sectional area necessary for any arm depends upon the means provided for preventing a sidewise stress on the lever, due to rotation of piston or friction of its connection to piston rod. The spool or cross head on the piston rod should be so designed that no sidewise strain can be thus produced in the lever.

b. The levers must have a double or bifurcated end at cross head.

The double end is less likely than a single end to put undue stress

on the lever as the rod turns, and is also less likely to give trouble from lack of lubrication or from a loosening of any small parts, and has proved to be the most satisfactory arrangement.

18. *Valve-motion Stand.* *a.* The valve motion stand must be securely dowel-pinned to the yoke castings, to prevent any movement after once adjusted.

19. *Cushion Valves.* *a.* Cushion-release valves regulating the amount of cushion steam retained at ends of stroke must be provided.

b. The cushion release must be through an independent port, as shown

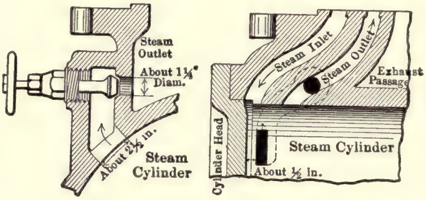


Fig. 372.—Cushion Valve.

in Fig. 372, so located as to positively retain a certain amount of cushion steam.

The old form of cushion release through bridge between ports is not acceptable. This form, while leading into the exhaust passage as formerly, differs by starting from a small independent port (about ½-inch wide by 2½ inches long) through the cylinder wall, located about ⅜ or ½ inch back from the cylinder head. (The exact position for affording the best action has to be determined by experiment with each different make of pump, as it depends somewhat on the extent of clearance space and on the point of closure of exhaust by piston and somewhat on the weight of reciprocating parts).

This style of cushion port makes the pump safer in case cushion valves are unskilfully left open too wide, and tends to prevent a pump from pounding itself to pieces in case of a sudden release of load, as by a break in suction or delivery mains, or by a temporary admission of air to suction pipe.

Pumps made with this form of cushion release have given very satisfactory results, and if the ports are properly located, there will be no rebound of piston.

Fig. 373.—Hand Wheel.

c. Cushion valves must be always provided with hand-wheels marked as per sketch, Fig. 373, for the reason that a very few men in charge of fire-pumps are found to clearly understand or to remember their use.

The lettering must be very open, clear and distinct, not liable to be obscured by grease and dirt, and of a permanent character.

It is desirable that spindle or wheel be so formed that a monkey wrench can get a grip to open a jammed valve. Fig. 374 shows the stem flattened for this purpose.

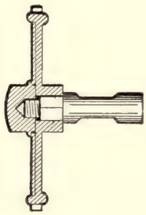


Fig. 374.—Flattened Stem.

d. The valve and stem of cushion valve must be in one piece without any swivel joint.

Swivel joints are apt to come apart and make it impossible to operate the valve.

20. *Piston Rods.* *a.* Piston rods for their entire length must be of solid Tobin Bronze, and the distinguishing brand of the manufacturers of this metal must be visible on at least one end of each rod.

b. The sizes must be not less than in table below.

Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
Diameter of rod................	2 inch	$2\frac{1}{4}$ inch	$2\frac{3}{8}$ inch	$2\frac{1}{2}$ inch

c. The size and form of connection of rod to piston plunger and cross head must be such that the stress in pounds per square inch at bottom of screw thread, or at such other point of reduced area as receives the highest tensile stress, shall not exceed 8000 pounds per square inch, when the steam pressure acting on the piston is 80 pounds per square inch.

d. Piston rod nuts, in both steam and water ends, must be tightly fitted, and preferably of a finer thread than the United States Standard. This is to

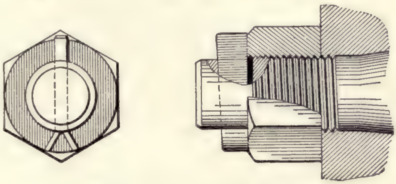


FIG. 375.—Lock Nut.

avoid as much as possible the unnecessary weakening of the rod at the bottom of the thread, and to reduce the tendency of the nut to work loose.

In practice eight threads per inch has been found to give good satisfaction.

e. In addition to a tightly fitting nut, some reliable device must be provided, in both steam and water ends, for absolutely preventing these nuts from working off.

Fig. 375 shows one form of such a locking device and illustrates the kind of security desired.

This device combines the advantage of a taper key and a split pin, and the elongated key-slot gives sufficient leeway to always insure that the key can be driven up tight against the nut and thus prevent it from even starting to work off. Other methods will be approved in writing, if found satisfactory.

21. *Valve Rods.* *a.* Valve rods for their entire length must be of solid Tobin Bronze, with sizes not less than in table below.

Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
Diameter of rod................	1 inch	$1\frac{1}{8}$ inch	$1\frac{1}{8}$ inch	$1\frac{1}{4}$ inch

b. The net area of valve-rod at its smallest section subject to tensile stress,

must not be smaller than at bottom of U. S. Standard screw thread on rod of diameter given above.

The construction of this rod as affecting lost motion at slide valve is specified under Art. 15.

22. *Stuffing Boxes.* *a.* All six stuffing boxes must be bushed at the bottom with a brass ring with suitable neck and flange, and the follower or gland must be either of solid brass, or be lined with a brass shell $\frac{3}{16}$-inch thick, having a flange next the packing, as shown in the sketch.

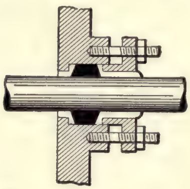


FIG. 376.—Stuffing Box.

The bottom of stuffing boxes and the end of the glands should taper slightly towards the center, as per sketch.

b. These glands should be strong enough to withstand considerable abuse, so as not to break from the unfair treatment of unskilled men.

23. *Pressure Gauge.* *a.* A pressure gauge of the Lane double tube spring pattern with 5-inch case must be provided and attached to the steam chest inside the throttle valve.

The dial of gauge should be scaled to indicate pressures up to 120 pounds and be marked "STEAM."

24. *Drain Cocks.* *a.* Four brass drain cocks, each with lever handle and of one-half inch bore, are to be provided, and located one on each end of each steam cylinder.

25. *Olling Devices.* *a.* A one-pint hand oil pump, to be connected below the throttle, and a one-pint sight feed lubricator, to be connected above the throttle, must be furnished with each pump.

b. Oiling holes must be provided for all valve motion pins, and for each end of both rock shafts.

26. *Stroke Gauge.* *a.* A length-of-stroke index must be provided for each side of pump. These must be of simple form for at all times rendering obvious the exact length of stroke which each piston is making, and thus calling attention to improper adjustments of cushion valves or stuffing boxes.

b. The gauge piece over which the index slides must have deep, conspicuous marks at ends of nominal stroke, and also light marks at extreme positions; it need contain no other graduations.

c. This stroke index must be rigidly secured to cross head in such a way that it cannot get loose or out of adjustment.

THE WATER END.

27. *Water Cylinders.* *a.* These must be of hard close iron with metal so distributed as to ensure sound castings, and freedom from shrink cracks.

b. The design should be along lines best calculated to resist internal pressures so as to avoid as much as possible the need of ribs for stiffening.

c. They must be capable of withstanding, without showing signs of weakness, the pressures and shocks due to running under the conditions mentioned in chapter "Tests for Acceptance," Arts. 48-54.

The suction chamber should be able to withstand a water pressure of 100 pounds.

Although suction chambers are not regularly subject to a pressure, it is sometimes desired to connect them to public water supplies, and where foot valves are used there is a chance of getting pressure on the suction, so that ample strength is necessary.

Foundry finish may be permitted on the joints at water cylinder heads and at hand-hole plates, provided surfaces are so true that a rubber packing not over $\frac{1}{16}$ of an inch in thickness is sufficient to secure perfect tightness.

d. Conveniently placed hand-holes of liberal size must be provided for the ready examination and renewal of valve parts at the yoke end of water cylinders and in the delivery chamber.

This will necessitate holes not less than 6x8 inches, or its equivalent, for the two largest-size pumps, and holes proportionately as large for the 500- and 750-gallon pumps. The easy access to the valve parts is of vital importance, and must receive careful attention.

e. The thickness of metal for cylinder shell, valve decks, partitions, ribs, etc., will depend largely upon the form of construction, but, in a general way, to establish safe minimums for the average water cylinder, of nearly cylindrical form, whose flat surfaces are stiffly ribbed, we submit the table below:

Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
	Inches	Inches	Inches	Inches
Thickness of cylinder shell when of nearly cylindrical form	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$
Thickness of valve decks when well ribbed	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
Thickness transverse partition, depending on ribbing	$1\frac{1}{4}$ to $1\frac{1}{2}$	$1\frac{1}{4}$ to $1\frac{1}{2}$	$1\frac{1}{2}$ to 2	$1\frac{1}{2}$ to 2
Thickness of longitudinal partition, depending on ribbing	$1\frac{1}{4}$ to $1\frac{1}{2}$	$1\frac{1}{4}$ to $1\frac{1}{2}$	$1\frac{1}{4}$ to 2	$1\frac{1}{2}$ to 2
Thickness of ribs	$\frac{3}{4}$	$\frac{7}{8}$	1	1
Thickness of suction chamber	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$
Thickness of delivery chamber	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$

Lighter construction than herein specified will not be regarded as satisfactory, and any construction will be finally passed upon on examination of drawings.

f. The bolting of all parts of the water end is to be of such strength that the maximum stress at bottom of screw thread will not exceed 10,000 pounds per square inch (disregarding for the moment the initial stress due setting up nuts) for a water pressure of 200 pounds per square inch, computed on an area out to center line of bolts.

No stud or tap bolt smaller than $\frac{3}{4}$ inch should be used to assemble parts subject to the stress of water pressure, as smaller bolts are likely to be twisted off. This does not apply to standard flanges where through bolts are used.

Although these pumps are not expected to be designed for a regular

working water pressure of 240 or 320 lbs., it is expected that bolts, shells, rods, etc., will be figured to stand this comparatively quiet, temporary, high pressure, exclusive of further allowance for initial stress due setting up of bolts, with a factor of safety of at least four.

This high test pressure is analogous to the custom of proving all common cast-iron water pipes to 300 lbs. and all common lap-welded steam pipes to 500 lbs. per square inch, and common water-works gate valves to 400 lbs., even though these are to be regularly used at much less pressure.

We are assured that castings no heavier than at present used by the best makers will stand this test, *if properly shaped and liberally bolted.*

g. For requirements for stuffing boxes, see Art. 22.

28. *Water Plungers and Bushings.* *a.* The "inside plunger and bushing" is preferred for all situations where the water is free from grit or mud.

b. Water-plungers must be of solid brass or bronze, and the bushing in which they slide must also be of brass or bronze. The composition of the plunger and its bushing should be of very hard, though dissimilar alloys, to ensure good wearing qualities.

For material and size of piston rods and lock for nuts, see Art. 20.

With poor alignment or bad workmanship or lack of skill in mixing the alloys, brass plungers are liable to score and give trouble; but with proper selection of alloys and true cylinders accurately aligned, they can be made to run all right wherever iron ones can. It is quite a fine point to get these wearing surfaces just right; and *this is wherein the experience, skill, and shop practice of one maker is likely to be much superior to that of another working under the same specification.*

c. The length of machined cylindrical bearing within the partition must be not less than two inches. The plunger bushings must have a faced seat transverse to its axis against partition, forming a water-tight ground joint not less than one-half inch wide.

Any rubber gasket or other compressible packing for making this joint water-tight is not acceptable.

d. The construction of bushing and hole in partition must be such that a cylindrical shell for use with a packed piston can be interchangeably inserted in its place and secured by the same bolts.

This can readily be arranged, and enables a packed piston to be inserted in place of a plunger subsequent to the installation of the pump with a minimum of expense, should this become desirable from change of conditions at any future time.

e. Small transverse grooves cut within the sliding surface of the plunger bushing, with a view to lessen the leakage, are not acceptable.

Although a slight advantage in this respect for clean water, they are a disadvantage on the whole, as dirt catches in them in the ordinary situation and cuts the plungers.

29 *Standard Dimensions of Plungers and Plunger Bushings.* *a.* To bring all these expensive parts to the same standard of weight and bearing surface, the following dimensions are specified as the least that will be acceptable. These are based on a length of plunger which uncovers the bushings one inch at end of nominal stroke:

SOLID BRONZE PLUNGERS AND BUSHINGS

Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
Plunger—	Inches.	Inches.	Inches.	Inches.
Diameter	7 or $7\frac{1}{4}$	9	10 or $10\frac{1}{4}$	12
Length	17	17	18	24
Thickness of transverse partition	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$
Thickness next to partition	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$
Thickness next to end	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
Number of ribs	4	4	6	6
Thickness of ribs	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$
Bushing—				
Length	7	7	8	10
Thickness at end	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
Thence tapered evenly to a thickness next to bearing of not less than	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$
Thickness at the center bearing not less than	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$

30. *Water Pistons and Bushings.* *a.* The "water piston with fibrous packing" is preferred for many situations in the West or South, or for water containing grit or mud, like that of the Ohio River; and, for the comparatively few cases where pump pressure governors are used, the packed piston will give better service and longer wear.

b. The removable bushing or cylinder in which this piston works must be of solid bronze.

c. As stated in Art. 28 *d*, this bushing should be so constructed as to be readily interchangeable with the bushing of the inside plunger type.

d. The length of cylindrical bushing must be such that the outer edge of packing will come short of the edge of bushing at contact stroke about $\frac{1}{2}$ inch and not uncover.

e. The thickness of the cylindrical bushings must be not less than that given in the following table:

BUSHINGS FOR PACKED WATER PISTONS

Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
Solid Bronze—	Inches.	Inches.	Inches.	Inches.
Thickness at extreme end	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$
Tapered evenly from end to a thickness next to bearing of not less than	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$
Thickness at center bearing, at least	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$

f. In other respects, the specifications for plunger bushings, already given in Art. 28, will apply to the above.

g. The water piston used in the shell described above must expose not less than 2 inches in width of fibrous packing, and must be of bronze, with disc and follower accurately turned to a sliding fit, so that the leakage past it will be a minimum, even when no fibrous packing is in place. There must be at least 2 inches in length of metallic bearing on both disc and follower.

The follower must be accurately centred, and fitted to hub of piston, so that alignment will not be disturbed if taken apart.

h. The water piston must be of simple and strong construction, with follower bolts tightly fitted, and with fibrous packing so cut as to prevent by-passing.

i. All materials used in construction of piston, except packing, must be brass, bronze, or other non-corrosive metal.

j. Bushing studs must be of Tobin bronze, and of such size and number, that the maximum stress at the bottom of the screw thread shall not exceed 10,000 pounds per square inch, in the event of plunger becoming fast in the bushing with 80 pounds of steam in the steam cylinders.

k. For each bushing stud there must be provided a composition nut and check nut.

l. All minor parts exposed to the action of water in water cylinder, that are not herein specified, must be of brass, bronze, or other non-corrosive material.

31. *Pump Valves. a*. All the suction and discharge valves in any one pump must be of the same size and interchangeable.

b. There must be a clear space around each rubber valve, between it and the nearest valve, equal to at least one-fourth of the diameter of the valve, or between it and the wall of the chamber of at least one-eighth of the diameter of the valve.

c. These valves must be of the very best quality of rubber, of medium temper with a face as soft as good wearing quality will permit.

They must be double-faced, so they can be reversed when one face is worn.

The quality of rubber is almost impossible of determination by brief inspection or by chemical analysis. The relative amount of pure gum and of cheaper composition may vary, or good material may be injured by defective vulcanization. The only safe way to secure excellence and uniformity is for the pump manufacturer to test samples of each new lot under severe duty (as by a week's run in a small special pump, with say 150 pounds pressure and heavy water hammer, or by some equivalent means) and to furthermore require the rubber manufacturer to mould a date mark as "(Name of pump manufacturer, lot 201—April 3, 1904)" on the edge of every valve, by which the pump manufacturer can keep track of those which prove defective.

32. *Size and Number of Pump-Valves. a*. The diameter of the disc of rubber forming the valve must not be greater than 4 inches or less than 3 inches.

Three and five-eighths inches diameter appears to be the size best meeting all the conditions, and has been adopted by several manufacturers but is not insisted upon.

There is some confusion between different shops about designating size of valves. The practice is here adopted, which is much the most widely used, of naming the diameter of the disc of rubber which covers the ports, and it is hereby specified that this shall be about one-half inch greater than the diameter of the valve-port circle which it covers, thus affording about one-fourth inch over-lap or bearing for the rubber disc all around its edge.

If valves are larger than four-inch there is an increased tendency to valve-slam at the very high speed at which the pump is designed to run, and if valves are smaller than three inches diameter the greater number tends to unnecessary multiplication of parts, and the ports being so small are a little more liable to become obstructed by rubbish.

b. The thickness of the rubber valve must in no cases be less than $\frac{5}{8}$-inch.

33. *Suction Valve Area.* *a.* The total lift of suction valves must not exceed $\frac{1}{2}$-inch.

b. The net suction valve port area and the total suction valve outlet area under valves lifted $\frac{1}{2}$ inch high must not be smaller than the figures given in the table below:

(1) Length of Stroke (in inches).	(2) Greatest No. Revolutions per Minute.	(3) Corresponding Piston Travel per Minute.	Approx. Actual Max. Piston Velocity at Full Speed per Column (3)x2.2.		(6) Net Suction Valve-port Area Regarded Necessary for this Speed Per Cent of Plunger Area.	(7) Total Suction Valve-outlet Area Under Valves Lifted $\frac{1}{2}$ Inch High.	(8) Discharge Valve Area.
			(4) Feet per Min.	(5) Feet per Sec.			
12	70	140 ft.	308	5.1	56%	56%	$\frac{2}{3}$ of Suction Valve Area.
16	60	160 ft.	352	5.9	64%	64%	

By "valve-outlet area," we mean the vertical cylindrical surface over the outer edge of the valve ports, i.e., the distance L multiplied by the circumference at the outer edge of the valve ports C. Thus for a four-inch valve, with ports inscribed in a three and one-half-inch circle, whose circumference is $3.5 \times 3.1416 = 11$ inches; the valve "outlet area" for one-half-inch lift would be $5\frac{1}{2}$ inches.

The actual velocity of piston during the middle portion of stroke is from 2.0 to 2.4 (average 2.2) times as great as the piston travel per minute (as determined in experiments by Mr. J. R. Freeman on several duplex pumps of different manufacture). This is because each piston stands still nearly half the time, or while its mate is working, and, moreover, moves more slowly near start and finish of stroke. The words "piston speed" are commonly incorrectly used, and refer to "piston travel." A clear understanding that the actual piston speed is *more than twice as great* leads to more generous valve design.

Large aggregate valve areas are necessary for pumps designed to run as fast as these, and experience has shown that to prevent valve slam at high speed and to accommodate high suction lifts, it is just as important to have a large "valve outlet area" as to have a large area of valve port.

It is valve slam or water hammer which commonly limits the highest speed at which a pump can be run. This water hammer may originate from the pulsations in a long or small suction pipe. The vacuum chamber

lessens it, but there is commonly some point of high water in the vacuum chamber that will give much smoother action than any other.

Valve slam in this style of pump is caused chiefly by the short rebound of the steam piston against the elastic steam cushion at the end of the stroke. This in turn snaps the valves down with a jump when the speed is high. Dividing this impact or slam on numerous valves of low lift, tends to break up and lessen the shock, therefore with valves of the size and style used in fire pumps, other things being equal, the less they have to rise and drop to let the water through them, the less will be the valve slam. This height of rise and drop is governed by the circumference rather than the port area. Experience and practice have shown that a ½-inch limit of lift is reasonable and does insure a smooth working pump under all ordinary conditions.

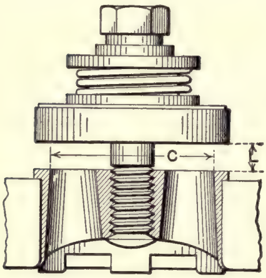


FIG. 377.—Water Valve.

c. The following table gives minimums for aggregate valve port area and aggregate valve outlet area for the different size plungers, figured on a basis of 56 per cent of plunger area for a 12-inch stroke, and 64 per cent for a 16-inch stroke.

	Size of Pump.	500 gal.	750 gal.	1000 gal.	1500 gal.
1	Diameter of plunger. Inches....	7¼″	9″	10″	12″
2	Area of plunger in sq. inches.....	41.28	63.62	78.54	113.10
3	56% of plunger area, or minimum aggregate valve-port area allowed per section. Sq. inches..	23.11	35.63	43.98	64% = 72.38
4	Minimum aggregate valve-port circumference, allowed per section. Inches..................	46.22	71.26	87.96	144.76
5	Minimum aggregate valve outlet area allowed per section for valves lifted ½ inch high. Sq. inches........................	23.11	35.63	43.98	72.38

d. If we consider using any one of the three sizes of valves below, whose port areas may be assumed approximately as given, then the necessary number of valves per section will be as in the table following:

Diam. Valve.	Diam. of Valve-port Circum.	Circum. of V. C. Circle.	Valve-port Area (Net), Square Inches.
3″	2½″	7.85″	3.5
3⅝″	3⅛″	9.82″	5.1
4″	3½″	10.99″	6.3

Size of Pump.	500 gal			750 gal.			1000 gal.			1500 gal.		
Size of valves, inches........	3	$3\frac{5}{8}$	4	3	$3\frac{5}{8}$	4	3	$3\frac{5}{8}$	4	3	$3\frac{5}{8}$	4
Necessary number of valves to satisfy (4) under *c*.....	6	5	6	9	8	7	11	9	8	19	14	14
Necessary number of valves to satisfy (3) under *c*......	7	5	4	10	8	6	13	9	7	21	15	12

The exact number and size of valves will, however, not be insisted upon provided the aggregate valve area and the aggregate valve outlet area for each section is not less than that given in the table under *c* for the limiting lift of $\frac{1}{2}$ inch.

Manufacturers will note that with the established lift of $\frac{1}{2}$-inch, the $3\frac{5}{8}$-inch valve will permit a valve outlet area more nearly equal to its port area than will either the 3-inch or 4-inch valves, and a *relatively* less number of valves will satisfy the specifications.

34. *Delivery Valves.* *a.* The total lift of delivery valves must not exceed $\frac{1}{2}$ inch.

This is to avoid valve slam, as explained in Art. 33.

b. The aggregate valve-port area should be restricted to about two-thirds the suction-valve area.

A small restriction of water-way through the delivery valves steadies the action of the pump and tends to prevent undue pulsations of pressure in the delivery pipe or fire hose. Fewer delivery valves than suction valves are, therefore, preferred, and if extra holes in the delivery deck are cast for shop purposes these had better be plugged than fitted with valves.

The suction valves require more generous port-circumference and port-area than delivery valves because when a pump has to suck its supply through a considerable height or through a long pipe there should be the least practicable waste of the atmospheric pressure in getting the water into the plunger chamber, or in retarding it from following the plunger in full contact. With the water once in the plunger chamber there is plenty of steam pressure available to force it out through the delivery valves.

35. *Valve Springs, Guards and Covers.* *a.* All valve springs must be of the best spring brass wire, and must be coiled on a cylindrical arbor.

Conical valve springs are not approved because the stress is not uniform throughout spring, thereby increasing the liability to breakage and the chance of their getting out of center and becoming "hooked up."

b. The valve spring must be held centrally at its top by resting in a groove in valve guard, substantially as shown in Fig. 378.

c. A light, rustless, metallic plate must be interposed between the bottom of the spring and the rubber valve, and must be the full area of the valve. This plate must also be formed with a raised bead to guide the spring at the bottom.

The weight of this plate should be small, for the inertia of the lifting

parts of the valves should be the least possible, to permit quick action and to avoid pounding.

d. For the average condition of a 10- or 15-foot lift, the stiffness of suction valve springs should be such that a force of about one pound per square inch of net port area will lift valve ¼-inch off its seat.

The springs on the delivery valves should ordinarily be from two to three times as stiff as just specified, but any other reasonable degree of stiffness which is proved to work well in practice will not be objected to.

For suction under a head, the greater snap with which water enters the plunger chamber when thus pushed in by say twice the atmospheric pressure renders it difficult to avoid water hammer at high speed. Extra stiff suction valve-springs will commonly aid in controlling this and should be used wherever pumps are to work under a head.

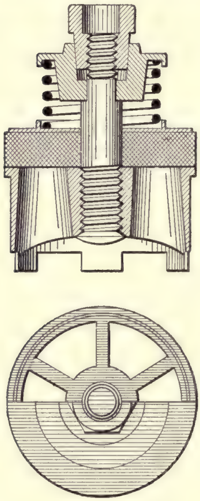


Fig. 378.—Valve.

An approved type of indicator water gate on the suction pipe near the pump, which can be partly closed, will enable the pump to run quietly at high speed. Such a gate is an extra not included in price of the pump.

36. *Sticking of Valves.* *a.* Steam fire-pumps should be started to limber them up *at least* once a week.

Although vulcanized India rubber is much the best material yet used for fire-pump valves, unfortunately the brass is sometimes corroded by the free sulphur contained in the rubber, so that if the pump is left standing for several weeks the rubber valve discs may become stuck to their brass seats, and, if suction has a high lift, there may not be vacuum enough to tear all the suction valves open when pump is started.

37. *Valve Seats.* *a.* All water valve seats must be of bronze composition. They may be either screwed into the deck on a taper or forced in on a smooth taper fit. With either arrangement, the seat must be either flanged out on the under side all the way round or be provided with a substantial lug opposite each rib, these lugs being expanded out after the valve is inserted.

If the valve seats are not expanded after being put in place, there is a possibility that now and then a valve seat will work loose and come out, thus crippling the pump.

b. The under side of the valve deck must be rounded over to give good bearing for the expanded part of the seat.

c. Three-inch valves must have four or five ribs, 3½-inch valves five or six ribs, and 4-inch valves six ribs.

Enough ribs must be provided to give proper support to the rubber valve, but too many are objectionable, as small ports would be liable to obstruction by refuse.

d. The edges of the valve-seat ports must be moderately rounded over to remove such sharp edges and points as would be liable to cut or damage the rubber valve when under pressure.

38. *Valve Stems.* *a.* All valve stems must be of $\frac{5}{8}$-inch Tobin bronze and of the fixed type, and must have the guard fastened on by one of the methods shown by Figs. 378 and 379.

Other methods may be approved, in writing, if found by test and experience to have especial merit.

b. These stems must be screwed into the seats on a straight, tightly fitting thread, and the lower end then well headed over into a countersink. The valve guard and nut must be of composition.

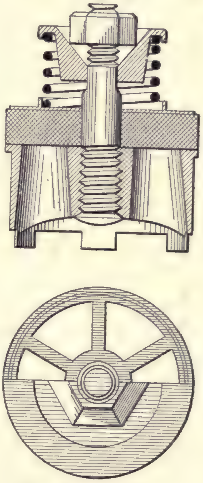


Fig. 379.—Valve.

In Fig. 378 the upper part of the stem is slabbed off on two opposite sides and fits a corresponding hole in the guard.

The guard, therefore, cannot turn. The outside of the special nut is fitted on a taper to the inside of the guard, and the nut tapped out to fit the five-eighths U. S. thread on the stem.

The action of the valve, whether with the spring or without, tends to drive these taper fits together, producing a frictional lock similar to that of a friction clutch; and although the nut may be loose on the thread, it cannot possibly work off.

It will be apparent that the taper fit on the nut must be so made as to always bear on the taper fit in the guard, and not bottom in the guard.

It is believed that with the present screw machine practice in shops of to-day these small parts can readily be turned out accurately and cheaply in large quantities. The nuts and guards made in any one shop must be exactly of standard dimensions, so that the product of different periods will be interchangeable.

The taper should be about one inch to one foot. With this taper the nut can be readily turned in or out, but there is friction enough to hold the guard and nut together even if the spring is off.

In Fig. 379 the top of the guard is recessed in the form of a hollow inverted pyramid of six sides, to correspond to a hexagonal nut. The angle of two opposite sides of this recess, which should be about 75 degrees, will both surely lock the nut and still permit of its being turned with a wrench.

The guard is kept from turning by slabbing off the stem in the same manner as described and shown in Fig. 378.

To facilitate the removal of the nut, the edges should be slightly chamfered. An unfinished nut simply drilled and tapped is all that is desired. Any hexagonal or square nut within the size of the tapered recess will be locked.

With this construction, the nut cannot turn in either direction without compressing the spring and is therefore locked, and, in the event of the spring breaking or being left off, the nut is well protected in its recess from the possible turning effects of water currents, and experiments have shown that it will stay in place.

With machine molding it will be possible to make these guards complete in foundry, requiring no machine work further than a possible broaching out of hole to fit the stem, as a fairly good fit is necessary.

While both of these devices are effective even though not tightened down to a shoulder, they should be so tightened for greater safety and to fix the lift at the half-inch limit.

39. *Pipe Sizes.* *a.* Water and steam pipe connections must have standard flanges to connect with pipes of the sizes given below.

Size of Pump. Gal. per Min.	Diameter of Suction Pipe. Inches.	Diameter of Discharge Pipe. Inches.	Steam Pipe.	Exhaust Pipe.
500	8	6	3	4
750	10	7 or 8*	$3\frac{1}{2}$	4
1000	12	8	4	5
1500	14	10	5	6

*Eight-inch preferred, this being the more common size for valves, fittings and pipes.

These suction pipe-sizes, although larger than common for trade pumps of the same size, are believed to be amply justified by experience, and exert a powerful influence toward enabling the pump to run smoothly at high speed with water cylinders filling perfectly at each stroke. No defect is more common than restricted suction pipes.

b. A single suction entrance at the end of the pump is to be provided unless otherwise specified by the purchaser.

Some situations render desirable side suction entrances, for permitting drafting water from two different sources of supply. These additional openings are to be considered as extras. Ordinarily, the purchaser can provide for such situations by proper piping at the single end suction entrance.

If there is to be but one suction opening on casting, this had best be at center, for the reason that, if suction pipe ever gets to leaking air, this air stands a better chance of being distributed equally to the two plungers, and has less tendency to make the pump run unevenly.

c. Standard flanges and standard bolt layouts as adopted by the Master Steam Fitters, July 18, 1894, must be used on all the above pipe connections, as per table given below.

SCHEDULE OF STANDARD FLANGES

Size of Pipe × Diam. of Flange. Inches.	Diameter of Bolt Circle. Inches.	Number of Bolts.	Size of Bolts. Inches.	Flange Thickness at Edge. Inches.
$3 \times 7\frac{1}{2}$	6	4	$\frac{5}{8} \times 2\frac{1}{2}$	$\frac{13}{16}$
$3\frac{1}{2} \times 8\frac{1}{2}$	7	4	$\frac{5}{8} \times 2\frac{1}{2}$	$\frac{7}{8}$
4×9	$7\frac{1}{2}$	4	$\frac{3}{4} \times 2\frac{3}{4}$	$\frac{15}{16}$
$4\frac{1}{2} \times 9\frac{1}{4}$	$7\frac{3}{4}$	8	$\frac{3}{4} \times 3$	$\frac{15}{16}$
5×10	$8\frac{1}{2}$	8	$\frac{3}{4} \times 3$	$\frac{15}{16}$
6×11	$9\frac{1}{2}$	8	$\frac{3}{4} \times 3$	1
$7 \times 12\frac{1}{2}$	$10\frac{3}{4}$	8	$\frac{3}{4} \times 3\frac{1}{4}$	$1\frac{1}{16}$
$8 \times 13\frac{1}{2}$	$11\frac{3}{4}$	8	$\frac{3}{4} \times 3\frac{1}{2}$	$1\frac{1}{8}$
9×15	$13\frac{1}{4}$	12	$\frac{3}{4} \times 3\frac{1}{2}$	$1\frac{1}{8}$
10×16	$14\frac{1}{4}$	12	$\frac{7}{8} \times 3\frac{5}{8}$	$1\frac{3}{16}$
12×19	17	12	$\frac{7}{8} \times 3\frac{3}{4}$	$1\frac{1}{4}$
14×21	$18\frac{3}{4}$	12	$1 \times 4\frac{1}{4}$	$1\frac{3}{8}$

Do not drill bolt holes on center line, but symmetrically each side of it.

On steam and exhaust openings loose flanges threaded for wrought-iron pipe must be provided.

Where the situation will not permit of a standard flange on exhaust opening for lack of room, a special flange threaded to fit the proper size wrought-iron pipe may be used.

40. *Air and Vacuum Chambers.* *a.* Air and vacuum chambers in accordance with the sizes given in the following table must be provided with all pumps. If the air chamber is cast iron, the pump manufacturers must warrant that it has been subjected to a hydraulic test of 400 pounds per square inch before it is connected to pump.

It is to be thoroughly painted inside and out to diminish its porosity.

SIZE OF VACUUM AND AIR CHAMBERS

	Vacuum Chamber is to Contain	Air Chamber is to Contain
500 gallon pump	13 gallons.	17 gallons.
750 " "	18 "	25 "
1000 " "	24 "	30 "
1500 " "	30 "	40 "

The air chamber, combined with connections for discharge pipe, relief valve, and hose valves, should be carefully designed to make the whole weight as small as possible. Keeping this weight down makes the pump run steadier and brings less stress on the flanges at high speeds.

An air chamber of hammered copper and warranted tested under a hydraulic pressure not less than 300 pounds per square inch is a little better than cast iron, as it holds air better, and being lighter it wrenches and strains the pump less when running fast and shaking, but because it costs from $25 to $50 more than cast iron, it is not often adopted.

b. The vacuum chamber must be attached to the pump in the most direct way practicable, but provision must be made for attaching it in such manner as not to prevent readily taking off the cylinder heads.

c. Every vacuum chamber should be provided on one side near the top with a ¼-inch pipe tap plugged. This to be used for attaching a vacuum gauge if desired.

41. *Pressure Gauge.* *a.* A pressure gauge of the Lane double tube spring pattern with 5-inch case, must be provided with the pump, and connected near to inboard side of air chamber, as shown in Fig. 381, by a ¼-inch cock, with lever handle.

The dial of this gauge should be scaled to indicate pressures up to 240 pounds, and be marked "Water."

This kind of gauge is used on locomotives and is the best for withstanding the vibration which causes fire-pump gauges to be often unreliable. Moreover, this double spring form is safer against freezing.

42. *Hose Valves.* *a.* Hose valves must be attached to the pump (and included in its price) as follows:

For the 2 stream or	500-gal. pump,	2 hose valves.
For the 3 stream or	750-gal. pump,	3 hose valves.
For the 4 stream or	1000-gal. pump,	4 hose valves.
For the 6 stream or	1500-gal. pump,	6 hose valves.

These are to be 2½-inch straightaway brass valves, without cap, and similar and equal in quality to those made by the Chapman Valve Company, the Ludlow Valve Company, or the Lunkenheimer Company.

The hose-screw at end of these valves is to be fitted to a hose coupling furnished by the customer, or where this cannot be procured may be left with the thread uncut.

To accommodate locations where all the lines of hose must lead off from one side of the pump—makers can furnish a spool piece or special casting to which the hose valves can be attached—but this is an extra not included in the regular price.

43. *Safety Valve.* *a.* A safety or relief valve of the Ashton, Crosby, American, or other make agreed upon in writing with this office, is to be regularly included in the price, and is to be attached to each pump; preferably extending horizontally inboard from base of air chamber, as shown in Fig. 381, so that its hand-wheel for regulating pressure is within easy reach. This hand-wheel must be marked very conspicuously, as shown in sketch.

FIG. 380.—Hand Wheel.

b. This valve is to be set ordinarily at a working pressure of 100 pounds to the square inch, and is to be of such capacity that when set at 100 pounds it can pass all the water discharged by the pump at full speed, at a pump pressure not exceeding 125 pounds per square inch.

For 500-gallon pump, a 3-inch valve.
For 750-gallon pump, a 3½-inch valve.
For 1000-gallon pump, a 4-inch valve.
For 1500-gallon pump, a 5-inch valve.

The relief valve must discharge in a vertical downward direction into a cone or funnel secured to the outlet of the valve. (See Art. 44.)

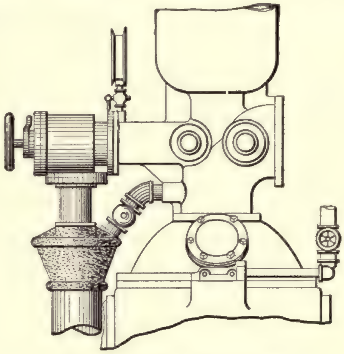


FIG. 381.—Safety Valve.

The valve must be so attached to the delivery elbow and discharge cone by flange connections as to permit of its ready removal for repairs without disturbing the waste piping.

44. *Discharge Cone.* *a.* This cone should be so constructed that the pump operator can easily see any water wasting through the relief valve, and its passages should be of such design and size as to avoid splashing water over into the pump room.

b. The cone must be provided with an opening to receive the air vent pipe

required by Art. 45, and the arrangement must be such that the pump operator can easily tell whether water is coming from the air pipe or is wasting through the relief valve.

c. The cone should be piped to some point outside of the pump house where water can be wasted freely, the waste pipes being as below.

Size of Pump.	Diameter of Waste Pipe from Cone.
500 gallon.........	5 inches.
750 "	6 "
1000 "	7 "
1500 "	8 "

The waste pipe can pass down to floor between the yokes at middle of pump. It should be piped in such a way that steam and gases from other drains or waste pipes will not work back through it, and, by being troublesome in the pump room, suggest the covering of the cone in any way, as it is desirable that the pump operator should *always* be able to see instantly any waste from the relief valve or air vent.

This cast-iron cone, connected to the safety valve and air vent, is included in price of pump, but the waste pipe beyond it is not.

45. *Air Valve.* *a.* An air vent with a brass gate valve and brass pipe for connecting up must be provided and connected with delivery elbow and discharge cone.

b. The size of this air vent should be 1 inch for 500-gallon and 750-gallon pumps, and 1¼ inches for the 1000-gallon and 1500-gallon sizes.

c. The hand-wheel of this valve must be marked as per sketch. The lettering must be very open, clear, and distinct, not liable to be obstructed by grease and dirt, and of a permanent character.

FIG. 382.—Valve Handle.

The object of this valve is to reduce the pressure above force valves and secure a prompt riddance of all air that may come through the water cylinders when first starting up.

This valve, of course, should be closed, when once pump is under way, to prevent waste of water.

46. *Priming.* *a.* Each pump must be fitted with a set of brass priming pipes and valves, according to either one or the other of the following methods:

b. For 1000 and 1500-gallon pumps, the priming pipes must be 1¼ inch. For the 500 and 750-gallon pumps, the pipes must be 1 inch. Pump-makers are to furnish these pipes and the fittings called for below, and are to connect them up providing a 2-inch outlet, looking upwards, ready for the supply from the priming tank.

The pipe from the priming tank to this outlet should be at least two-inch, and may be of iron, and is to be furnished by the purchaser. All parts furnished by the pump-maker are to be of brass, and are to be included in the price of the pump.

Controllable Valve Arrangement. *c.* Four two-seat controllable valves, one for each pulsation chamber, and of the general type illustrated in Fig. 383,

must be provided. In these the inlet of water and the outlet of air are simultaneously opened and closed by the pump operator.

This valve can preferably be provided with a flange connection in place of the threaded one, and secured to water cylinder with three five-eighth bolts. This will permit of easier fitting up as to pipe con-

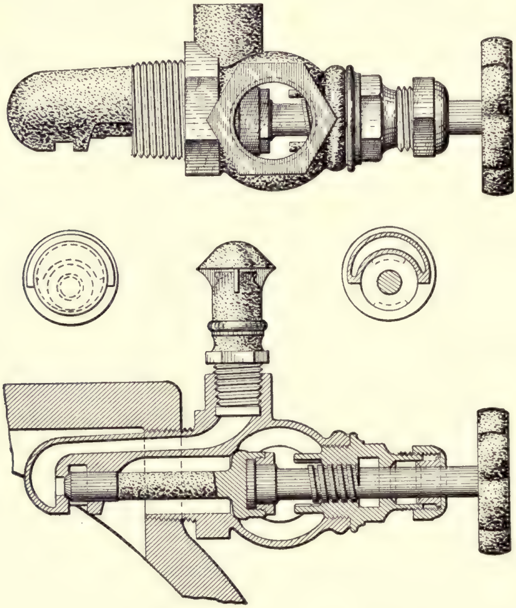


FIG. 383.—Valves.

nections. Objection has been raised to this double-seated valve from the possible difficulty of keeping both seats tight. If desired, the stem between the two seats may be somewhat enlarged and provided with a suitable spring, thus giving flexibility between the two seats and preventing all trouble from uneven wear.

d. The hand-wheel of each of these valves must be marked as per sketch so that the pump operator may clearly understand their use. The lettering must be very open, clear, and distinct, not liable to be obscured by grease and dirt, and of a permanent character.

FIG. 384.—Valve Handle.

e. There must be provided and fitted to each combined valve a check and umbrella-top air vent, as shown in Fig. 385. This fitting must have a clear passageway through it, the full equivalent of a half-inch bore.

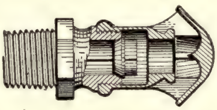


FIG. 385.—Check Valve.

The check valve is to permit the outflow of air, but to prevent the influx when the plunger is sucking.

This method is preferred to the one using rubber priming checks, as now and then a rubber valve will stick on its seat and thus prevent priming of one of the chambers. In this arrangement the pump operator has absolute control over the priming water into each chamber.

Another advantage is that the connection of the air vent with the priming valve ensures that the air vents will be opened; and further, by the vigorous spurting out of water as soon as the pump is primed, the pump operator is reminded that the priming valve should be closed.

Should the pump operator, however, through a mistaken idea of the proper method of operation, think that the priming should be continued until all air was exhausted from the suction pipe, and the pump running in normal condition, there would be some by-passing between chambers, but as there is a free vent for the air, the main result would be simply to limit the amount of air exhausted per stroke, from the main suction, by the amount of water which entered a chamber in this way. The amount of water thus entering, however, would not be appreciably greater than that which would enter from the priming tank with the check-valve arrangement.

If, in spite of the warning given by the spurting air vents, the pump operator should neglect to close the priming valves when the pump was running normally, the priming tank would eventually be overflowed; but this would not be as serious as the drawing in of air from an exhausted priming tank, which would result with the check-valve method, were the main two-inch valve similarly neglected.

Rubber Check Valves. *f.* Four rubber check valves, one for each pulsation chamber, and similar to ordinary pump valves, must be provided. The chambers for these should preferably be made as a part of the pump cylinder, thus securing a compact arrangement.

g. The valve seat should have three ribs to the central hub, supporting the rubber valve. The net port area through the valve should not be less than one and one-half square inches.

This valve seat should rest in an inverted position, and can be so fitted up as to be readily removed. The valve stems can be of the removable type screwing into the seat, but must be made long enough to receive a check nut on the opposite side of seat. This will effectually lock the stem in place.

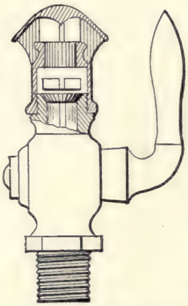


FIG. 386.—Check Valve and Air Cock.

h. Care must be taken to arrange the water passages through and about these priming checks, so as to avoid all air pockets and so as to reduce to a minimum the possibility of the valves becoming choked up by refuse.

i. The valve seats, stems and all parts must be of composition and of strong rugged design, so fitted up that there is the least chance for the rubber valves to stick, and with all parts securely put together the valves must be readily accessible.

j. The valve springs must have only sufficient strength to keep the valves on their seats, so that they will freely open even with the low head of priming water often existing.

k. There must be provided, and attached to the top of each plunger chamber, a brass check valve and air cock with umbrella top, as shown by Fig.

386. This cock and valve must have a clear passageway through them—the full equivalent of a half-inch bore.

The check valve is to permit the outflow of air, but to prevent the influx when the plunger is sucking. Cocks with lever handles are used, as these show clearly whether they are open or shut.

l. There must also be provided a two-inch brass gate valve for the general control of the water to the four check valves. The hand-wheel of this valve must be marked as per Fig. 384. The lettering must be very clear, open, and distinct, not liable to be obscured by grease, and of a permanent character.

It is essential for a properly working pump that the main two-inch priming valve should be closed as soon as the pump is primed. Otherwise water will be drawn from the priming tank, lessening the lifting power of the pump through the main suction, and if this is continued the priming tank will often be exhausted and air drawn into the pump, interfering with its proper action. It is for this reason that the marking on the priming valve is required.

For all average situations, either method of priming permits of getting the pump under way in a very few minutes, but, for cases where the suction pipe is over 300 or 400 feet in length, or sometimes where the lift is over 18 feet, or where there is a combination of long length and lift within these limits, so much time is consumed in exhausting the air from the suction pipe that it becomes desirable to supplement this method.

For such situations, a steam ejector connected to the suction pipe near the pump is advised, and may be required in addition to the regular priming pipes and tank. The size of the ejector should be sufficient to exhaust the suction pipe within about three minutes. Such ejectors will be considered as extras not included in the ordinary pump fittings.

For cases where pump can only take its suction under a head, if absolutely certain that the level of the suction water will never fall below level of center of pump, these priming pipes may be omitted, but openings for them, into the pump shell must be provided and capped or plugged.

A foot valve on a fire-pump suction is not advised except in very rare cases, as with a lift of 18 feet or a suction pipe 500 feet or more long. A foot valve is not needed when there is a good efficient set of priming arrangements, as described above, and it is commonly found it gives a false sense of security, and that with a fire pump left standing several days the water will often be found to have leaked back, so that it is no better than if no foot valve had been used.

A foot valve must of necessity generally be located where it is inaccessible for quick repairs, and as they grow old, foot valves are often a source of trouble. Where a suction pipe is exposed even slightly to frost, a foot valve is especially objectionable.

A priming tank is provided by the purchaser in all cases where there is ever to be any lift on the suction. It is generally advised that this tank have a capacity of one-half of what the pump can throw at full speed in a minute. This means 250 gallons for a 500-gallon pump, and 500 gallons for a 1000-gallon pump, etc. It is the intention to make the pump a truly " independent source " of supply, therefore the need of a special priming tank.

Older Priming Arrangements. The form of priming arrangements heretofore used, with metal check valves, one main two-inch priming valve, and one-inch priming pipes, separate controllable air cocks, may be retained on all pumps at present in service, and will be considered satisfactory, if kept in good order.

If in any case such checks give trouble the priming arrangement may be changed and valves like Fig. 383 or rubber checks, as described in sections *f–j*, made up in detachable form, may be put on if desired, where the connections on the pump permit them.

Where neither method is desired or where neither is feasible the faulty checks may be replaced by a special type such as are now made for this use by the Locke Regulator Company, of Salem, Mass. These are one-inch check valves, adapted to use a small disc of medium hard rubber, similar to a pump valve.

These fittings are very near the dimensions of the commercial check valve, so that with sight shortening of piping connections they will fit into the present arrangements, and give satisfaction.

47. *Drain Cocks. a.* Five brass drain cocks, each with a lever handle and of half-inch bore, are to be provided, and located one on each end of each water cylinder, and one above the upper valve deck.

Care should be taken to select a pattern of cock whose passageway is the practical equivalent of a half-inch hole. Some patterns of half-inch commercial cocks although threaded for half-inch pipe thread have but a quarter-inch hole through them. Such are not acceptable.

Tests for Acceptance.

48. *Test for Smoothness of Action. a.* Provide outlets for the water; start the pump slowly, gradually open steam-throttle to bring the pump to full speed. The pump should run smoothly at the rated full speed of 70 revolutions per minute (or 60 revolutions if a 1500-gallon pump) with full length of stroke, and meanwhile maintain a water pressure of 100 pounds per square inch.

If the hose lines are short, or discharge is too free, partly close the water outlet valves, thus throwing an extra back pressure on the pump equivalent to that which would be produced through a greater length of hose.

During this trial it is preferable to discharge the water through lines of 2½-inch cotton rubber-lined hose, preferably each 150 feet long, each connected directly to the hose outlets on the pump, and each line having a 1⅛-inch smooth nozzle at its outer end. Two lines should be connected for a 500-gallon pump, three for a 750, and so on, having as many lines as rating of pump requires.

A hose line 150 feet long, with an inside surface of average smoothness, and with a 1⅛-inch nozzle attached, will require about 80 pounds pressure at the pump to discharge 250 gallons per minute, and the nozzle pressure will be about 45 pounds. Therefore, with lines attached as above, a pressure at the pump of about 80 pounds should represent a discharge about equal to the rated capacity of the pump, and would ordinarily correspond with the rated full speed revolutions.

If the pump runs smoothly under these conditions, it is well to open the throttle somewhat further, and bring the pressure at the pump up to 100 pounds. This will give a discharge of about 280 gallons per stream, or about 12 per cent in excess of the rated capacity. The revolutions will, of course, correspondingly increase, and under all ordinary conditions a pump should run smoothly at this higher capacity, though a little more vibration and pounding would be expected than when running simply at its rated speed.

After cushion valves are adjusted there should be no noteworthy water hammer or valve-slam. Sometimes valve-slam is not the fault of the pump, but arises from an obstructed suction pipe. It is objectionable to doctor water hammer in a pump by shifting air into the suction, as this cuts down the efficiency and is a poor expedient.

The quietness of that part of the hose near the pump, or its freedom from rubbing back and forth crosswise an inch or more with each pulsation of the pump, is a good index of the pump maker's skill in securing uniform delivery. Bad pulsation quickly wears holes in the hose, and to reveal this is the object of testing *with hose connected directly to the pump.*

49. *Test of the Internal Friction.* *a.* This is shown by the reading of steam gauge compared with water pressure gauge at air chamber.

Tests have generally run about as follows, for pumps running at full rated speed:

Size Gallons per Minute Capacity.	Ratio of Steam Piston Area to Water Piston Area.	Water Pressure lbs. per Sq.in.	Steam Pressure Theoretically Necessary, Disregarding Friction.	Excess of Steam Pressure Needed to Overcome Friction Back Pressure, etc.	Actual Steam Pressure Found Necessary at the Pump.
500	4 Times	100	25	15	40
750	3 "	100	33	12	45
1000	3 "	100	33	12	45
1500	$2\frac{3}{4}$ "	100	36.5	13.5	50

b. The steam pressure needed will vary slightly with the freedom of the exhaust pipe and with the tightness of the packings, etc., but a steam pressure of 45 pounds at the steam chest should suffice for 100 pounds water pressure on pump in proper adjustment.

50. *Test of Strength and Tightness.* *a.* First, shut the main valve between the pump and the fire system lest a sprinkler head be burst, then shut all water outlets nearly, but not quite tight, so pump will move very slowly. Screw safety valve down hard. Slowly and carefully admit steam pressure sufficient to give 240 pounds per square inch water pressure.

b. With this extreme pressure all joints should remain substantially tight, and the slow motion of the pump should be tolerably smooth and uniform. (The leakage of a few drops here and there and a little unsteadiness of motion are to be expected.)

c. If boiler pressure is above 85 pounds, the safety valve on pump should be attached and screwed down only enough to hold the required pressure. For with 100 pounds or more of steam the water pressure might be carried too high.

After completing the above test slack off on safety valve, setting it so that it will begin to open at about 100 pounds pressure.

51. *Test of Capacity of Safety Valve.* *a.* The relief valve may next be tested by first adjusting it to pop at 100 pounds, then shut the main outlet to pump, and then shut the hose gates one by one, and thus force all the discharge through the relief valve, meanwhile opening steam throttle, so as to

run pump *first at two-thirds speed or about fifty revolutions per minute,* and finally at full speed (seventy revolutions). The safety valve (relief valve) should carry all this and not let the pressure rise above 125 pounds.

The pressure in a quick-moving fire-pump necessarily fluctuates 5 to 15 pounds at different points in stroke, and an air chamber of reasonable size cannot wholly remove this. Therefore the safety valve must be set at about 15 pounds higher than the intended average working pressure; otherwise it will get to jumping with almost every stroke.

52. *Test of Internal Leakage or Slip.* *a.* Set safety valves at 115 pounds, shut all water outlets, admit steam enough to give 100 pounds water pressure, then pump will move very slowly under the influence of the leakage past plungers; about one revolution of pump per minute shows a proper accuracy of fit. Anywhere from one-third to two revolutions per minute is satisfactory.

Too tight a fit is bad, as if not exceedingly uniform it induces scoring or fretting of the metals. Moreover, should pump happen to be run dry for a few minutes before catching its suction a slight warming and expansion of the plunger may cause it to stick and fret.

53. *Test With Maximum Working Pressure.* *a.* For this, alternately shut down the main outlet gate and adjust the hand-wheel of the safety valve, and open up on the throttle as may be required, running pump at say one-half speed (or, in experienced hands, at full rated speed), and note the greatest water pressure which the full boiler pressure (unless boiler pressure is above 85 pounds) will yield with pump at full speed.

Sometimes it may be necessary to force water through very long lines of hose, or to an unusual height.

Steam fire engines are not infrequently called on to give 200 pounds per square inch water pressure.

To test short hose lines with anywhere near so high a pump-pressure is dangerous, lest nozzle kick and pull itself away from the man holding it and thresh around; but the ability of the pump may be tested by putting this high-pressure delivery mainly through the safety valve, or in part through the partially closed main outlet gate.

It is not advisable to carry this water pressure above 200 pounds in this test at the factory, although in the shop test the water pressure is carried to 240 pounds, and engine driver should stand with his hand on the throttle.

54. *Test for Maximum Delivery.* *a.* This can best be tried by adding one or, in some cases, two more streams than the pump is rated to deliver by attaching the extra lines of hose to some hydrant near, and then speed up the pump gradually, to see how fast it may be run before violent pounding or slamming of valves begins.

Sometimes the increased delivery can be drawn off through an open hydrant-butt, meanwhile holding sufficient back pressure to show 100 pounds on the water gauge by partly closing the discharge valve.

The engine driver should stand with his hand on or near the throttle when thus speeding the pump.

It is all right to run a fire-pump up to the utmost speed possible before water hammer begins, and very often a pump, while new and if favorably set up, can deliver 25 to 50 per cent more than rated capacity: nevertheless, although expert treatment can force 1000 gallons from a 16x9x12 pump we can rate it as only a 750-gallon pump. *There must be some margin to allow for wear and for the possible absence of the expert at time of fire.*

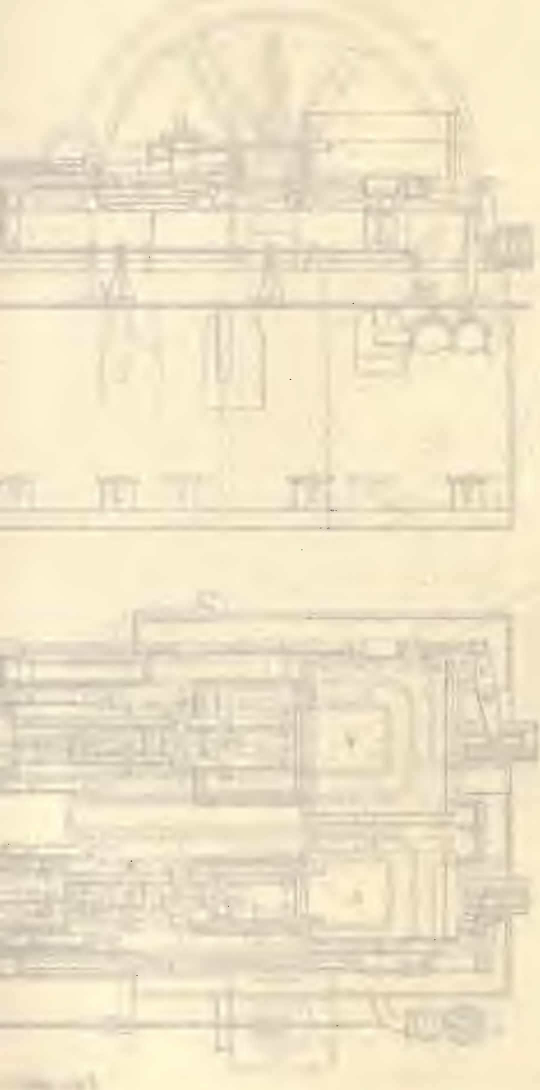
The main points of difference between the "National Standard" and the "Trade Pump" are:

Brass plungers instead of cast-iron plungers.
Wrought-iron side levers instead of cast-iron.
Bronze piston rods and valve rods instead of iron or steel.
Pump has brass lined stuffing boxes instead of cast-iron.
Rock shafts are brass bushed.
Area of water valves is 25 to 50 per cent greater.
Steam and exhaust passages 20 to 50 per cent greater.
Suction pipe connections two or four inches greater diameter.
Cushion valves better arranged.
Air chamber is made much larger.
Shells and bolting are warranted especially strong.

The following necessary fittings are included in the price, and regularly furnished as a part of this pump, viz:

A capacity plate.
A stroke gauge.
A vacuum chamber.
Two best quality pressure gauges.
A water relief valve of large capacity.
A cast-iron relief valve discharge cone.
A set of brass priming pipes and special priming valves.
From two to six hose valves.
A sight feed cylinder lubricator connected above throttle.
A one-pint hand oil pump connected below throttle.

The **hydraulic pressure pumps** are built for very high pressure. In these pumps, as shown in Fig. 387, the large steam cylinders are connected with small water cylinders *GG*. In the pump shown steam is used in the cylinder *E* and then discharged into cylinder *F*. The piston rods *AA* are connected to the cross head *B* and from this the plungers extend into the two water cylinders. The connecting rod *C* is forked at *DD* so that it clears the right hand pump *G*. The use of the fly-wheel permits of the expansive use of steam. Such pumps are often connected to a cylinder closed by a weighted ram (an accumulator), so that the water may be stored under pressure which may amount to several thousand pounds per square inch. The use of an accumulator permits the pump to run continuously even though the use of water is intermittent. When there is such a load that the pump will have a number of periods of inaction, a direct-acting pump is better than a fly-wheel pump with a fixed stroke, as there is then little or no



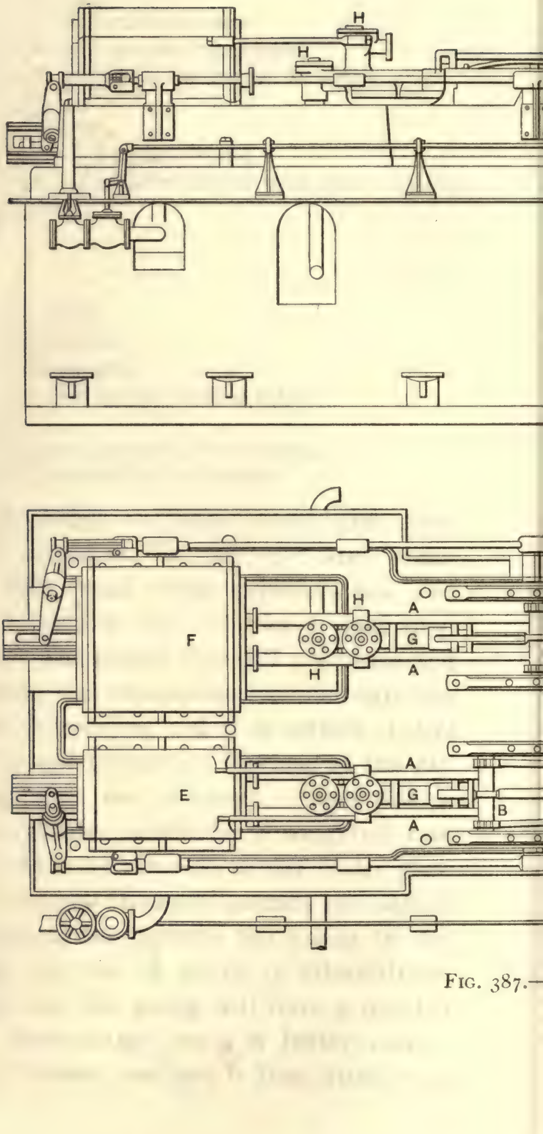
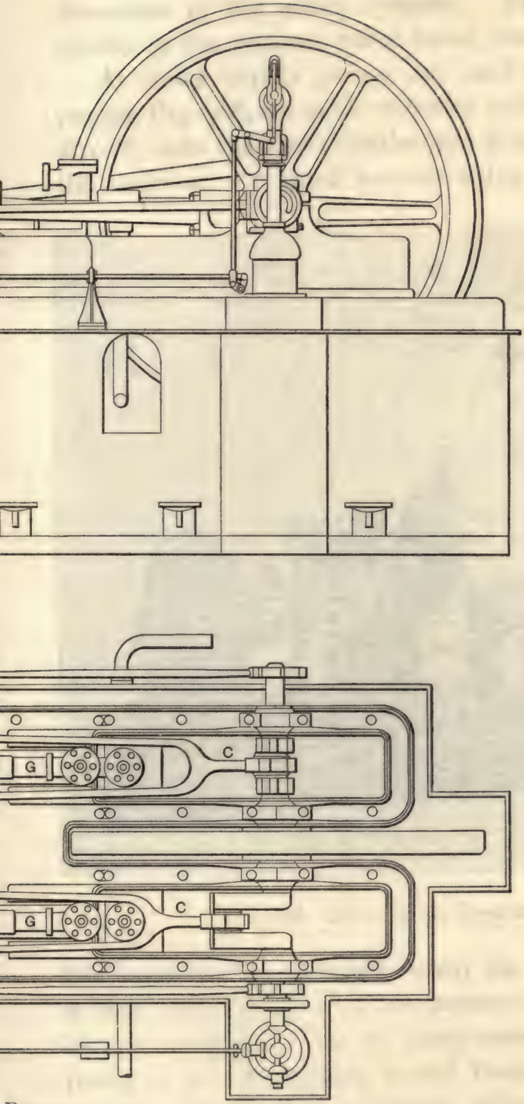


Fig. 387.—



Pump.

(To face page 496)

danger of the pump being wrecked from a collection of condensation in the steam cylinder. The valve boxes *HH* are similar to the pressure valve boxes shown earlier.

At times triplex pumps are used for this purpose. Such pumps, Fig. 388, are made of heavy parts. The water is pumped directly into the ram cylinder and from this it is discharged to the reservoir. A small by-pass valve is set to discharge from

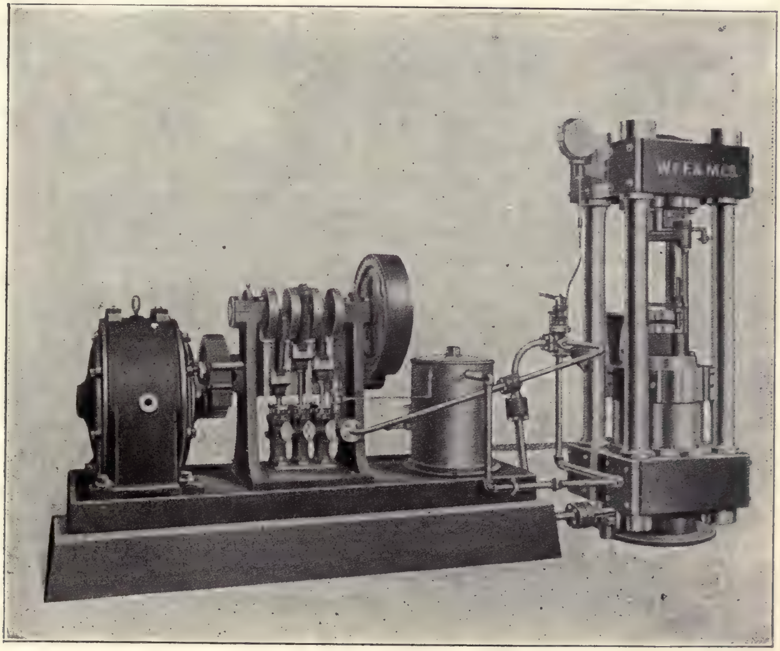


FIG. 388.—500-ton Press and Pump.

the pump to the reservoir, when the hand valve at the press is shut off or as soon as the pressure in the system reaches a certain amount. The 15 horse-power motor used with this pump of the Waterbury Farrel Foundry and Machine Co. is directly connected to the pump, although at times belted connections are used. The proportions of the pump and press are given in the table below, which is taken from the catalogue of the builders.

Press.			Pump.		
Capacity	tons	500	Diameter of plunger	in.	1⅛
Diameter of ram	in.	16	Stroke	"	6
Maximum stroke of ram	"	5	Approximate capacity, cubic inches		1060
Diameter top of ram	"	22	Pressure, lbs. per sq.in		4975
Top platen to ram when down	"	40	R.P.M. of crank-shaft		61½
Opening between rods	"	18½ by 12½	Ratio of gearing		13.67 to 1
Floor to top of ram	"	26	Floor space	in.	91 by 32
Size of supply pipe	"	1½xx			
Floor space, with pump	"	165 by 36			
Extreme height from floor	"	82½			
Total weight, including motor	lbs.	17175			

Figs. 389 and 390 show two **air pumps for beer racking.** These pumps are mounted on the air reservoir which acts as a

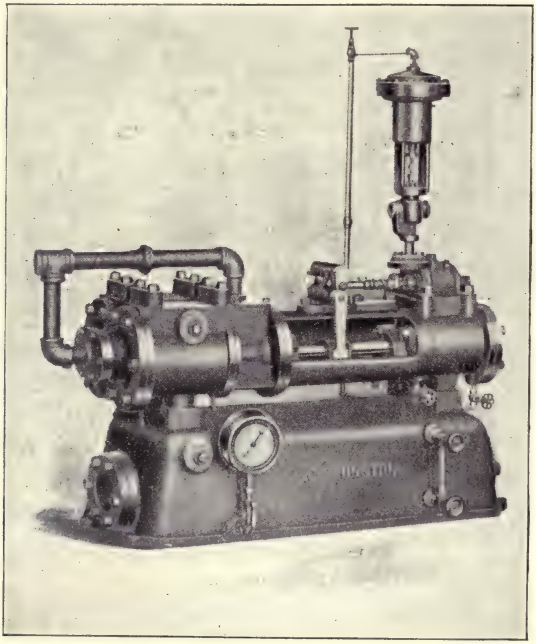


FIG. 389.—Worthington Beer-racking Pump. (Size 5¼×6×5.)

bed plate. This gives proper storage capacity. The cylinders are lined with a composition. The regulator shown on each pump is connected with the air reservoir and keeps the pressure constant. The pump of Fig. 390 shows how a fly wheel can be used between the cylinders of the steam and air ends. A yoke is placed in the rod between the two cylinders and a crank pin of the shaft operates between these. This is the equivalent

of a connecting rod of infinite length, and by its use the fly wheel may be applied so that steam can be employed expansively.

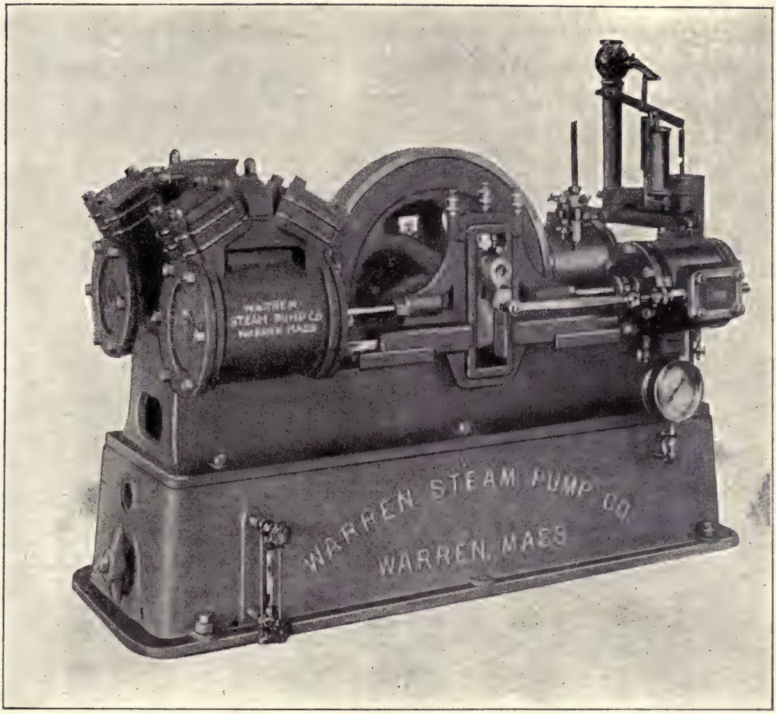


FIG. 390.—Beer-racking Air Pump.

The table below gives the sizes of this type of pump as made by Worthington:

Diameter of Steam Cylinders.	Diameter of Air Cylinders.	Length of Stroke.	Annual Capacity of Brewery.	Sizes of Pipes for Short Lengths to be Increased as Length Increases.				Approximate Space Occupied. Feet and inches.			
				Steam Pipe.	Exhaust Pipe.	Suction Pipe.	Delivery Pipe.	Length.		Width.	
$5\frac{1}{4}$	$5\frac{7}{8}$	5	20,000 to 50,000 bbls.	$\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	1	3	7	1	9
6	8	6	50,000 to 100,000 bbls.	1	$1\frac{1}{4}$	2	$1\frac{1}{2}$	4	9	2	4
$7\frac{1}{2}$	9	10	100,000 to 150,000 bbls.	$1\frac{1}{2}$	2	3	3	6	7	2	5
9	12	10	150,000 to 200,000 bbls.	2	$2\frac{1}{2}$	4	3	6	10	3	0

To designate the sizes, give the diameters of the steam and air cylinders and the length of stroke.

The **hydraulic ram** in one of its modern forms is shown in Fig. 391, while Fig. 392 illustrates the method of connecting it. The drive pipe *F* connects the source of supply *A* with the ram at *B* at a lower level. The water can escape from the ram by an opening or waste valve *K*. The velocity set up at the discharge opening is sufficient to lift the valve *G* and suddenly

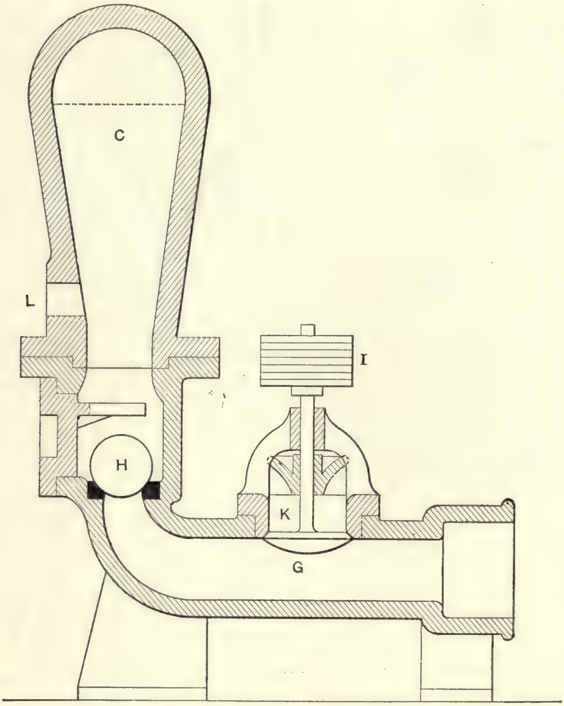


FIG. 391.—Hydraulic Ram.

close the passage. The momentum of the water in the pipe line produces an increase in the pressure near the valve, as shown from the rise from *a* to *b* in Fig. 393. This figure was drawn by the pencil of an indicator attached to the pipe *F*, known as the drive pipe, while the drum was moved uniformly. The heights, therefore, represent pressure while length represents time. The increase of pressure will cease as soon as the pressure is sufficiently great to open the ball valve *H* when the water

flows into C. The inertia and friction of the valve being overcome, the pressure drops with a backward wave to c and returns to d, after which the ball valve closes; the pressure drops in the space around G and a backward wave may even reduce this to a pressure below the atmosphere. In any case the weight I is

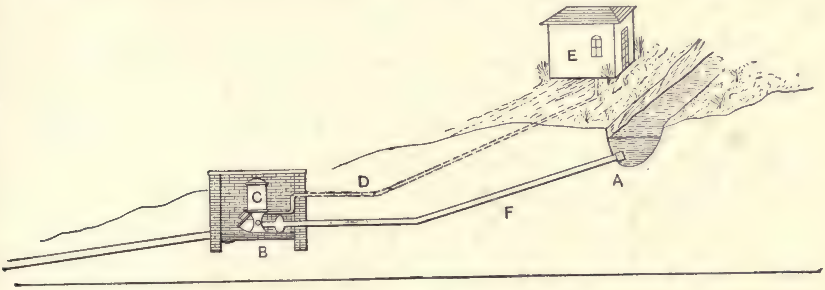


FIG. 392.—Arrangement of Ram.

sufficient to force G down against the water pressure due to the elevation of A, after which the operation is repeated at g, h, i, etc. The discharge is taken from L by the discharge pipe D which conducts it to the tank house.

In making and installing the ram, the design of pipe and size is fixed by practise, although there are theoretical deductions made at times. (Zeitschrift des Vereines Deutscher Ingenieure,

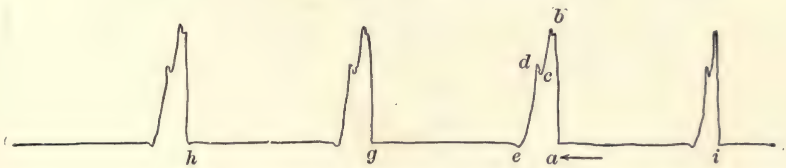


FIG. 393.—Card from Hydraulic Ram.

Jan. 15, 1910.) The drive pipe F should be of ample size and should be laid as straight as possible. It should be of a length at least equal to three-quarters the head against which it is to pump, or five times the head causing the flow through the drive pipe. The inclination of the drive pipe should never be over 30° at any place, and when the length or proper angle cannot be had it is well to coil the drive pipe to a large radius so

as to accomplish this. The length of the pipe is necessary to produce the momentum for driving. The length of delivery pipe should not be over twenty times the lift and when a longer pipe must be used less lift will be obtained.

The efficiency of the hydraulic ram is

$$E=\frac{w(H-h)}{Wh},$$

where

H = head from ram to reservoir in feet;
h = head from supply to ram in feet;
w = water pumped in lbs;
W = water discharged through waste valve in lbs.

This efficiency varies from 0.40 to 0.60, and by assuming 0.50 for its value, the equation may be used to find W when w, H and h are known. The drive pipe should then be made large enough to give a velocity of three feet per second and the delivery pipe two feet per second.

$$A_{\text{drive}}=\frac{(W+w)\times 144}{62.5\times t\times 3},$$

$$A_{\text{del.}}=\frac{w\times 144}{62.5\times t\times 2}.$$

CHAPTER XII

INJECTOR AND PULSOMETER

THE Sellers injector, Fig. 394, has been taken as a type of the injector. In starting, the handle A is drawn back a short distance permitting steam to enter the space B between the two tubes. The pressure of this steam between the two edges produces so high a velocity of discharge across the upper end of space C that it entrains the air at that point, driving it out through the openings D E into the space G, and from this

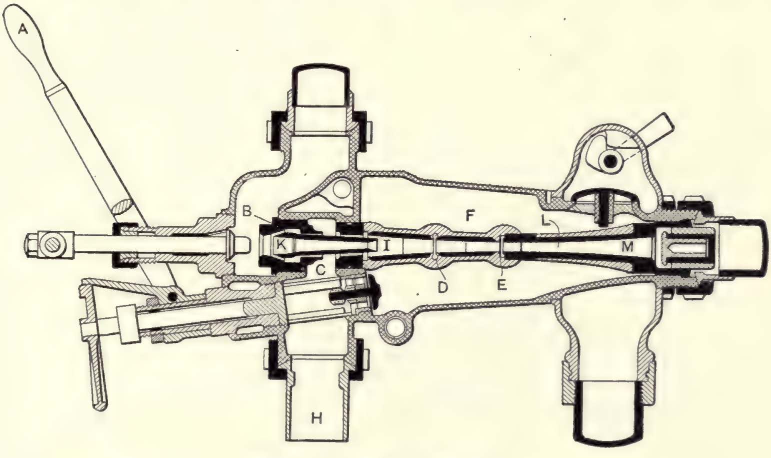


FIG. 394.—Sellers Injector.

into the atmosphere. This produces a partial vacuum at C and water enters through the suction pipe H. This water is taken up by the steam discharge, driven through the combining tube I and is afterwards discharged through D, and E, finally appearing at the overflow.

When water appears at the overflow, the handle A is drawn back completely and steam then discharges through the main nozzle K. The steam in this acquires a high velocity due to

the drop in pressure in the nozzle; this velocity is many times greater than that acquired by water under the same drop in pressure on account of the small density of the steam.

The steam is then condensed by the water, but when the steam particles draw together to form a drop of water their velocity is maintained and drops of water, moving with a very high velocity, result. This condensed steam can then strike the body of water drawn through H and by impact, impart to it a velocity greater than that which would be acquired by water discharging from the same boiler. If this is the case, this mixture of water and condensed steam could enter the boiler against steam pressure.

The combining tube I is made convergent since the steam is gradually condensed in the passage and the water gradually increases in velocity. When the throat L is reached the steam is condensed and the water is supposed to have its maximum velocity. From this point the delivery tube M diverges, making the velocity less, and with this comes a change of velocity head into pressure head until at the end there is sufficient pressure to force open the valve O when the mixture will enter the boiler.

The pressure in the space within the end of the combining tube reaches a point of low vacuum of about 25″ while at the end of the steam nozzle it is at atmospheric pressure. At the small part of the nozzle known also as the throat, the pressure is found to be 0.58 of the boiler pressure. The pressure at the throat of the delivery tube is usually about atmospheric pressure.

If now the pressure of the steam is p pounds per square inch, absolute, and the barometer reads p_a, the pressure against which the water must enter is $(p-p_a)$ and if Δ is the density of the water or weight per cubic foot at the throat, the head in feet of water to be overcome is

$$\frac{(p-p_a)144}{\Delta}=h_d,$$

Δ may be taken as 50 pounds for the first approximation. The reason for this low density is the fact that steam particles

are mixed with the water. Such a result has been found from experiment. The velocity at this point will be

$$V_d = \sqrt{2gh_d}.$$

The velocity of water drawn through the suction tube is produced by the pressure p_c within the combining tube, and if the water is lifted h feet and the friction is h_f feet the head causing flow is

$$\frac{144(p_a - p_c)}{62.5} - h - h_f = h_s;$$

the velocity of the water is

$$V_s = \sqrt{2gh_s}.$$

The steam discharging from the nozzle is under a pressure of p originally, and in the combining tube it is under a pressure of p_c. To find the velocity, the heat content H of the initial steam is found, considering the quality of the steam and then the heat content H_c at the lower pressure, assuming that the steam has the same entropy at the two points. If 15 per cent friction loss is assumed, the velocity V of the steam is given by the equation:

$$V = \sqrt{0.85 \frac{2g}{A}[H - H_c]} \quad \text{where} \quad H = q + xr.$$

Having now the velocities of the steam, water and mixture, the equation for impact may be used to find the amount of water per pound of steam.

$$V \times 1 + V_s M = (M + 1) V_d.$$

Since impact is not perfect, the sum of the first products has to be multiplied by a factor K. Mr. Strickland L. Kneass, in his "Theory of the Injector," has shown that a value of 0.50 should be used for K in the formula:

$$K(V \times 1 + V_s M) = (M + 1) V_d,$$

$$M = \frac{KV - V_d}{V_d - KV_s}.$$

With M known, the condition of the mixture can be found by equating the heat on each side of the throat in which q_d is the only unknown.

$$H+Mq_s=(M+1)q_d,$$

q_s = heat of liquid in suction,
q_d = heat of liquid of discharge,
H = total heat content of entering steam.

The density of the water in the discharge depends on the temperature of the water and the amount of steam left in the water at this point. Kneass has shown how this may be found, and in some cases its value is $0.21w$, where w is the weight of a cubic foot of water at the temperature of discharge. This result was determined for a small discharge. The value $0.75w$ is more nearly the value to be used in practice. This would give 45 pounds with water weighing 60 pounds per cubic foot.

If the weight of water wanted per hour is W pounds, the amount of steam per second is

$$\frac{W}{60\times 60M}=w',$$

hence the orifice must be of area sufficient to carry $w'(1+M)$ pounds of density Δ at the speed of V_d feet per second. The area at the throat is therefore

$$a_d=\frac{w'(1+M)}{\Delta\times V_d}\text{ sq.ft.}$$

The diameter of this expressed in millimeters gives the nominal size of the injectors according to some makers.

The area of the throat of the steam nozzle is sometimes made two or three times the area of the delivery throat, but this may be designed by considering the velocity set up by the fall of pressure from p to $0.58p$. Let H be the heat content for the steam under the conditions at entrance and H_t that at $0.58p$, but with the same entropy, then

$$V_t=\sqrt{0.98\frac{2g}{\Delta}(H-H_t)}.$$

The factor 0.98 is used here because of the short easy curve.

$$A_t=\frac{w'\times S_t}{V_t}.$$

S_t is the specific volume of the steam at the throat.

If H_e is the heat content of the steam at the end of the nozzle where the pressure has fallen to about atmospheric pressure, the following may be used to find the area at the end:

$$V_e=\sqrt{0.85\frac{2g}{A}(H-H_e)},$$

$$A_e=\frac{w'S_e}{V_e}.$$

H_e is taken from the same entropy column as H but in finding V_e, the volume is that which corresponds to a heat content $H'_e=H_e+0.15(H-H_e)$. The reason for this is the fact that the loss due to friction has been changed into heat and remains in the steam increasing the heat content over that resulting from true adiabatic expansion.

The nozzle is usually rounded to the throat area with quadrants and then the remainder of the nozzle is made along a straight line to the end. The combining tube slopes from the delivery throat to an area somewhat larger than the nozzle end.

The delivery tube should be made of gradual convergence, and Kneass recommends ("Theory of Injectors," p. 37) to use a curve such that the negative acceleration along the tube is constant. This gives the equation:

$$R=r\sqrt[4]{\frac{V^2}{V^2-2ay}}.$$

Where R is the radius of the tube at a distance y from the throat of radius r and a is the acceleration while V is the velocity at the throat.

Suppose that the velocity is to be destroyed in l feet of tube, then

$$c=\frac{V^2}{2\times l}.$$

This may be put in the above equation and the following results:

$$R = r\sqrt{\frac{1}{1-\frac{y}{l}}},$$

from this the various radii may be found along the tube.

The shapes of the tubes actually used in practice have been determined by experiment, and the above serves as a guide for certain leading dimensions.

The pulsometer described in Chapter II is built under a number of names and with various shapes, but of the same principle. Fig. 395 shows the Emerson Pump. The cylindrical vessels *AA* are for the same purpose as the pear-shaped vessels of the pulsometer.

The pulsometer is operated by the action of the steam on the water direct. The quantity of water to be handled by a given size is fixed from practice and the table below gives these sizes taken from a catalogue.

TABLE OF SIZES OF EMERSON STEAM PUMPS

Number.	Diameter of cylinders. Inches.	Length of Cylinders. Feet.	Size of Steam Pipe. Inches.	Size of Suction. Inches.	Size of Discharge. Inches.	Capacity in Gallons per Minute.	Capacity in Gallons per Hour.	Capacity in Gallons per Day, 24 Hours.	Dimensions Over all in Inches. Breadth.	Width.	Height.	Approximate Weights in lbs.
1	6	6	3/4	3	2 1/2	225	13,500	324,000	16 1/2	18	97 1/2	950
2	8	6 1/2	1	4	3	415	24,900	597,600	21 1/2	21	104	1370
3	10	7	1 1/4	5	4	725	43,500	1,044,000	26	24	113	1905
4	12	8	1 1/2	6	5	1200	72,000	1,728,000	29 1/2	27 1/2	127	3100
5	16	8	2	8	6	2100	126,000	3,029,000	43 1/4	33	132	4400
6	20	8	2 1/2	10	8	3275	196,500	4,716,000	51 1/2	36 3/4	135	5400
7	24	8	3	12	10	4700	282,000	6,768,000	....	....	....	7000

Capacities in gallons per minute, stated in table, vary with the steam pressure and height of lift. Special sizes above those listed, on application. Pumps made entirely of bronze, when called for, at special prices.

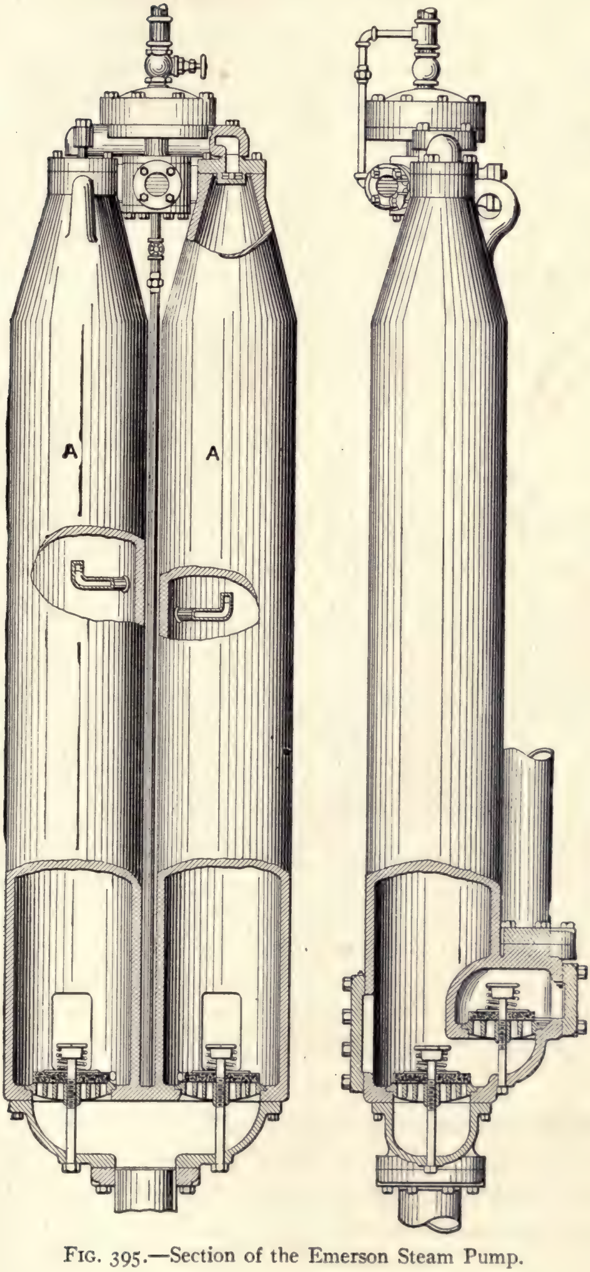


FIG. 395.—Section of the Emerson Steam Pump.

To compute the quantity of steam used, suppose that steam is supplied at a pressure p_1 and is used within the chamber at pressure p_2. Assume that there is no radiation and if this is the case the steam in passing through the valves is reduced in pressure but the heat content remains the same. Hence

$$H_1 = H_2.$$

From H_2 at the pressure of p_2, the specific volume S_2 of the steam can be obtained and then for a given quantity of water, V cu.ft., the weight of steam used would be

$$\frac{V}{S_2} = m.$$

The heat in each pound of this steam is $H = q_1 + x_1 r_1$, and if the temperature of discharge is t_d the heat chargeable per pound of steam is $H - q_d$.

q_d and the temperature of the mixture is given by the equation

$$Mq_s + m(q_1 + x_1 r_1) = (M + m)q_d + A\,144 p_2 (M + m)\sigma,$$

$$M = \text{weight of } V \text{ cu.ft. of water} = \frac{V}{62.5},$$

$$q_s = \text{heat of liquid in suction.}$$

$$\sigma = \text{volume of 1 lb. of water} = \frac{1}{62.5}.$$

The efficiency of the pump is

$$E = \frac{\text{work}}{\text{heat}} = \frac{144 p_2 V}{778 m (H - q_d)}.$$

The above equations have considered no loss due to initial condensation nor radiation, and these depend on many conditions. It may be said that the steam will probably be increased to $3m$ or more and this will reduce the efficiency to about one

third its former value. The term q_d will be changed as its equation becomes

$$Mq_s + 3m(q_1 + x_1 r_1) = (M + 3m)q + \text{Radiation} + \text{Work}.$$

Radiation $= K(A)(T_1 - T_2)t$;

$K = 300 =$ amount of heat radiated per sq.ft. per hour per degree difference in temperature;

$A =$ area of outside of vessels;

$T_1 =$ temperature on inside of pulsometer;

$= \dfrac{T_{\text{steam}} + T_d}{2}$

$T_2 =$ temperature air;

$t =$ time in hours in which V cu.ft. of water is pumped.

CHAPTER XIII

AIR LIFT PUMPS AND PNEUMATIC PUMPS

THE air lift pump, as was mentioned in Chapter II, was patented in 1880 by James P. Frizell, and Pohlé took out a

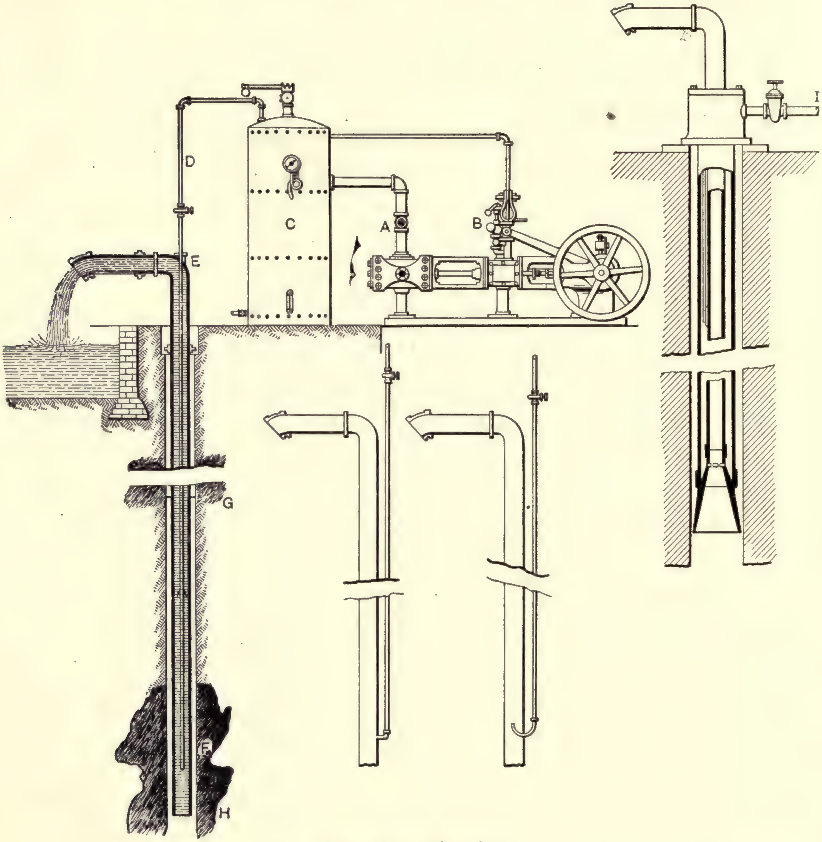


FIG. 396.—Air Lift.

patent in 1886, although these two are by no means the first records of this type of pump.

Fig. 396 shows the general arrangement of a plant for an

air lift. In this figure the air pump is attached to a storage tank from which the air is conducted to the well. The air compressor is shown to be of a single stage type although in many cases two stage compressors are used to increase the efficiency. The air cylinder *A* is in tandem with the steam cylinder *B*. The air from the storage tank *C*, with its gauge and safety valve, is conducted to the head of the discharge pipe at *E* through the pipe *D*. This pipe is continued down to a point *F* near the bottom of the delivery pipe. The well is usually cased outside of the delivery pipe through all earthy matter to solid rock if certain ground waters are to be kept from the well. The water in the well stands at a height h below the discharge and the depth of immersion from *G* to *H* is called h'.

The figure shows several methods of introducing the air. In the first place the pipe is carried down inside of the main pipe and opens directly into the delivery pipe. There are two other figures showing the small pipes carried down into the well and introduced into the side of the delivery pipe, while the fourth figure shows the method of carrying the air down in an annular space between two pipes, the inner one of which is the delivery pipe. A slide valve, controlled from the top of the well, admits air at some height above the bottom for the purpose of introducing air at a higher point when starting the apparatus. The air in this case is introduced at *I* into a head at the well top.

There are several methods of arranging the well tops for the reception of the discharge and for the introduction of the air pipes. These are shown in Fig. 397. In the first a concrete head *A* receives the discharge from the deflecting cap *G*. It is carried away from this by the conduit leading to the reservoir or irrigation ditch. The handle at *H* controls the admission of the air by a rod which extends to the lower end of the air pipe. When the water is to be carried to a higher level, the S bend shown at *B* is used, but the control valve entering the pipe at the ground level is similar to that used in *A*. When an elbow is used, the connectioon for the air pipe is shown at *C* and the forms at *D* and *E* are those used with the air

supplied in the annular space outside of the delivery pipe. In *E* the discharge is caught in a tank before being delivered, while in *D* the discharge is directed into the irrigation ditch or cistern.

Such installations may be arranged at considerable distance from a central air compressing station. One station is used to furnish air to a number of wells. The pipe lines should be arranged so that the air travels at a rate of from 2000 to 4000 feet per minute. The area of this pipe is usually one-sixth the area of the delivery pipe. Some pump makers claim that this method of raising water should not be applied when the water has to be raised over 80 feet, but others name 180 to 200 feet as the limit, and when the water is to be carried by the air

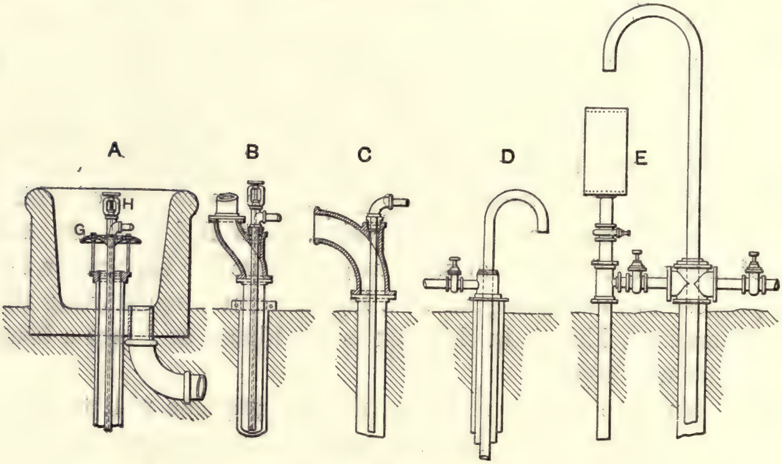


Fig. 397.—Air Lift Well Tops.

pressure in addition to a small lift the distance is limited to 700 or 800 feet. Be this as it may, greater lifts are used, although the cost of lifting may be excessive.

When greater lifts are required it might be well to use multiple lifts discharging through a portion of the height to a deep pipe reservoir in the main well and from this to another.

The compressors should in all cases be arranged to compress

the air with as nearly isothermal compression as possible. The only way to do this is to use an inter-cooler between the stages.

When the pressure is from 60 to 300 pounds, two stages should be used, and above this three or more stages. The air pipe should be introduced near the bottom of the discharge pipe and should be immersed so far that the ratio of h' to h is 3 to 1 at the start and 2.2 to 1 in operation, according to a test on a particular well to find out what depth of immersion gave the best results. This result, however, should not be used as a general law as there are many conditions affecting it. Some claim that the ratio should be 1.2:1 to 4:1. These will do as guides. The following table gives a series of tests showing what may be expected:

Place.	Immersion $\frac{h'}{h}$	Efficiencies.
Tunbridge, England	3:1 to 2.2:1	36%
Grinnell, Iowa	.....	29.6%
San Francisco	0.6:1	16% 43%
San Francisco	1:1	19% 42%
San Francisco	1.4:1	34% 41%
San Francicso	2.4:1	15% 24%
San Francisco	3.9:1	2%
——	1.5	50
——	1	40
——	0.66 to 1	30
——	0.50	25
——	0.43 to 1	20

The efficiency shown in this table is the ratio of the work done on the water to that done in compressing the air.

The pressure to be carried on the air system is greater than the water pressure at the lower end of the discharge pipe or $p=0.434h'+5$. This quantity diminishes after the pump starts to act as the head h' decreases, due to the removal of water. The difference between the head at start and while running, represents the head causing flow into the well. This quantity will vary in different localities, depending entirely on the formation and the nature of the water bearing rock.

The amount of free air varies according to different authors.

One gives the amount as 3.9 cubic feet to 4.2 cubic feet of free air per cubic foot of water, or

$$\text{cu.ft. of free air per min.} = \frac{LW}{16.824}.$$

L = lift of water above surface in well in feet.
W = cu.ft. of water per min.

A pump manufacturer recommends the following:

$$\text{cu.ft. air} = \frac{LW}{19},$$

while another uses:

$$\frac{LW}{15} = \text{cu.ft. of air per min.}$$

The air is transmitted at 2000 feet per minute, and the area of water pipe should be about six times that of the air pipe. The immersion should be $1\frac{1}{2}$ times the lift. The table below has been given in "The Engineer" for June 1, 1906:

Size of Well.	Water Pipe.	Air Pipe.	Gallons per Minute.
4″	$1\frac{1}{2}$	$\frac{3}{4}$	25
$4\frac{1}{2}$″	2	1	50
5″	$2\frac{1}{2}$	1	75
6″	3	$1\frac{1}{4}$	100
7″	$3\frac{1}{2}$	$1\frac{1}{2}$	150
8″	4	$1\frac{1}{2}$	200
9″	5	2	300
10″	6	2	450

This same article gives the following relation for the cubic feet of free air per cubic foot of water:

Lift.	Cu.ft. Air per Cu.ft. Water.	Submergence if 1.5.	Air Pressure in lbs.
25	2	37·5	17
50	2	75	33
75	4·5	112.5	49
100	6	150	65
125	7·5	187·5	82
150	9	225	98
175	10.5	202.5	115
200	12	300	130

As an example, the following data may be mentioned: A lift of 129 feet was obtained from a well 300 feet deep; the water was 44 feet from the top of the well, leaving an 85-foot lift above the well. The well was 8 inches in diameter and the discharge pipe was $3\frac{1}{2}$ inches with a $1\frac{1}{4}$-inch air pipe. The pump gave 82.5 gallons per minute and required 7.43 cubic feet of air per cubic foot of water. The air pressure required was 107 pounds and the loss in pressure was 9%.

As another example of the use of the air lift pump, the plant at Redlands, California will be mentioned. This was described in the "Engineering Record," Vol. 51, p. 8. The plant was built to replace machinery at four wells, from 450 to 570 feet deep, which were separated some distance. The first pump was of centrifugal form and was operated by a motor; the second, a plunger pump driven by a steam engine; the third, a centrifugal pump with a steam engine, and the fourth had not been used. It was decided to operate these wells by air from a central station. The wells were to be piped with 4-inch and 7-inch discharge pipes, extending down from 306 to 360 feet, while the air pipes were $1\frac{1}{4}$ to 2 inches in diameter.

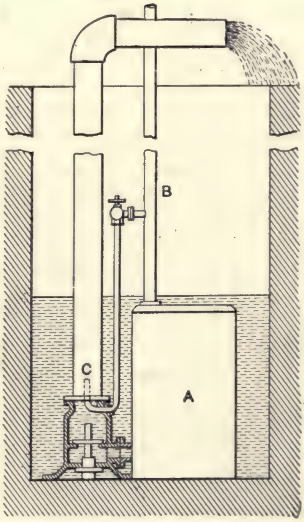


FIG. 398.—Wheeler System.

The station was a brick building 40×46 feet and equipped with return tubular boilers 66 inches in diameter and 16 feet long. These used oil for fuel. A 13 and 26×30 Cross compound engine with 14 and 22×30 two-stage air cylinders in tandem, supplied 1124 cubic feet of free air per minute to 125 pounds at 85 R.P.M.; at this time it developed 190 I.H.P. The boiler feed pumps at the station were a $2\frac{1}{2} \times 4$-inch triplex pump and a $4\frac{1}{2} \times 3 \times 4$-inch duplex pump.

The 48-hour test on this plant gave the following results:

	Mean for 48 Hours.	Maximum.	Minimum.
Boiler pressure	146.7	150	142
Air compressor pressure	90.98	94	85
Vacuum in inches	24.83	25.25	24.5
Well No. 1:			
Air pressure	87.4	89	84
Depth water level	101.44	109	98
Well No. 2:			
Air pressure	93.6	95	92
Depth water level	90.37	93	86
Well No. 3:			
Air pressure	87.03	90	85
Depth water level	110.67	116	99
Well No. 4:			
Air pressure	85.4	86	84
Depth water level	102.2	105	101
Fuel consumed in barrels per 24 hrs.	16.3	17.2	15.7
Rate of pumping in gals. per 24 hrs.	3,157,622	3,280,824	3,131,395

When sufficient immersion can not be had, the Wheeler system, Fig. 398, may be used. Water is admitted into one of the two vessels A by drawing out the air, when water will flow in from the well; then, on allowing air pressure to enter through B, this forms an equivalent head on the water.

The action of the pump and compressor will now be examined. Suppose that v cubic feet of air are admitted per cubic foot at the bottom of the discharge pipe, under a pressure of h' feet. The density of the mixture of 1 cubic foot of water and v cubic feet of air will be $\frac{1}{1+v}$, neglecting the relative weight of air. At any distance x from the top of the column $h+h'$, where the pressure varies from $h'+34$ at the bottom to 34 feet at the top, the volume v will be:

$$v_x = \frac{v(h'+34)}{\frac{x}{h+h_1}(h')+34} = \frac{(h+h_1)(h'+34)v}{h'x+34(h+h')},$$

and the density is $\frac{1}{1+v_x}$.

Now the weight of dx feet of water, of one square foot area, is

$$w\frac{1}{1+v_x}dx,$$

or the total pressure at the bottom of the well is

$$wh' = \int_0^{h+h'} w\frac{1}{1+v_x}dx$$

$$= w\int\frac{1}{1+\frac{(h+h_1)(h'+34)v}{h'x+34(h+h')}}dx$$

$$wh' = w\int\frac{h'x+34(h+h')}{h'x+34(h+h')+(h+h')(h'+34)v}dx$$

$$= w\int\frac{ax+b}{ax+b+c}dx$$

$$= w\left[\frac{ax+b+c}{a} - \frac{c}{a}\log(ax+b+c)\right]_0^{h+h'}$$

$$= w\left[\frac{a(h+h')}{a} - \frac{c}{a}\log\frac{a(h+h')+b+c}{b+c}\right]$$

$$wh' = w\left[h+h' - \frac{(h+h')(h'+34)v}{h'}\right.$$

$$\left.\log\frac{h'(h+h')+34(h+h')+(h+h')(h'+34)v}{34(h+h')+(h+h')(h'+34)v}\right]$$

$$h' = h+h' - \frac{(h+h')(h'+34)v}{h'}\log\frac{h'+34+(h'+34)v}{34+h'v+34v}$$

$$\frac{h'}{h+h'} = 1 - \frac{h'+34}{h'}v\log\frac{(h'+34)(1+v)}{34(1+v)+h'v}$$

$$\text{Rel. Immersion} = \frac{h'}{h+h'}.$$

If now, for a given depth h', different values of v be assumed

and $\frac{h'}{h+h'}$, determined, a curve may be plotted for the values of $\frac{h'}{h'+h}$, as ordinate, with v as abscissa, then for any given h the curve will give the volume v of compressed air associated with one cubic foot of water.

$$\frac{v(h'+34)}{34}=V_a,$$

the amount of free air to be associated with one cubic foot of water.

If water is desired to be delivered at a given rate, at the bottom of the main discharge, the velocity of entrance is given by the equation:

$$v=\frac{Q}{a}$$

Now $$v=\sqrt{2gh''},$$

or $$h''=\frac{v^2}{2g}.$$

If h'' is found, this amount should be subtracted from h' on the left hand side of equation for $\frac{h-h'}{h'}$. In that case, the head inside of the casing is h'' feet less than that outside, and this

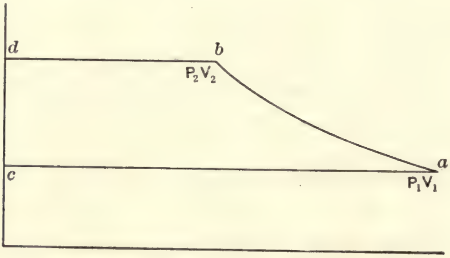


FIG. 399.—Compressor Diagram.

will give the velocity v of the discharge. This method gives the rational manner of designing.

To find the power and size of the compressor, consider Fig.

399, in which the curve of compression ab is of the form $pv^n = K$. It is desirable to have this curve a rectangular hyperbola, but that is impossible, as with water jackets around the cylinder the exponent is rarely below 1.2 and 1.4 is usually found. The work done in compression is

$$W = P_2V_2 + \int_{v_2}^{v_1} pdv - p_1V_1;$$

now

$$pV^n = K = p_1V_1^n = p_2V_2^n.$$

Hence

$$\int pdv = K\int V^{-n}dv = \frac{K}{1-n}(V_1^{1-n} - V_2^{1-n})$$

$$= \frac{p_1V_1 - p_2V_2}{1-n}.$$

The work becomes:

$$W = p_2V_2 + \frac{p_1V_1 - p_2V_2}{1-n} - p_1V_1$$

$$= \frac{n}{n-1}(p_2V_2 - p_1V_1) = \frac{n}{n-1}p_1V_1\left(\frac{p_2V_2}{p_1V_1} - 1\right)$$

$$= \frac{n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

This is the theoretical power, in which p_1 is the suction pressure and V_1 the volume taken in if there is no clearance or leakage. There is no effect of clearance on the work of a compressor as the work obtained on the expansion of the clearance air is equal to the work required to compress it. There is an effect due to leakage and this is to change the work by the factor $\frac{1}{\text{vol. eff.}}$. The volumetric efficiency is the ratio of the air delivered to the amount which should be delivered. This is about 95 per cent. The friction of the compressor increases the amount of work done, so that the net H.P. required to apply to the V_1 if this is the amount of free air required per minute is:

$$33{,}000 \times \text{H.P.} = \frac{1}{\text{mech. eff.}}\,\frac{1}{\text{vol. eff.}}\,\frac{n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right].$$

The clearance effects the displacement by cutting out the amount of air *ef*, Fig. 400, and caring for the amount *fa* only

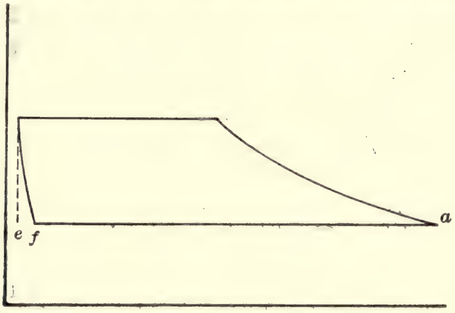


FIG. 400.—Compressor with Clearance.

The factor which gives this if $cl=$ per cent clearance is:

$$1-\frac{1}{cl}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}+\frac{1}{cl}=f.$$

The displacement then is

$$D=\frac{V}{\text{vol. eff.}\times(f)};$$

from the formula

$$D=2NLA$$

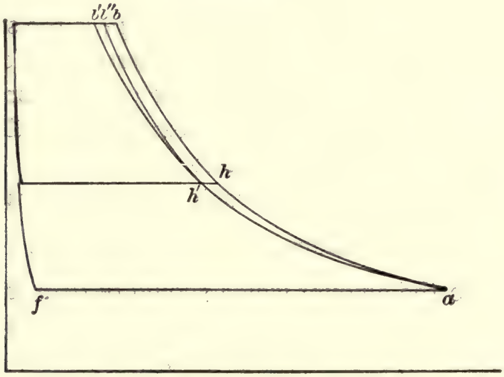


FIG. 401.—Multi-stage Compression.

for a double acting compressor the size of the cylinder and the number of revolutions may be found to give the necessary free air V. Allowance should be made for piston rods.

Since it is not possible to bring the curve of compression close enough to the isothermal, a method of multi-staging the compression has to be devised. In this, Fig. 401, the compression is carried to a pressure p_2' when the air is discharged into a series of tubes or around them, while on the other side of the tube surface cold water is circulated. In this manner the air can be left in contact a sufficient time to be brought to the original temperature, and when taken into the second cylinder at h' it is on an isothermal from a. In this way the work of the area $h\ bb''\ h'$ is saved. The work in this case is:

$$W=\frac{n}{n-1}\left\{p_1V_1\left[\left(\frac{p_2'}{p_1}\right)^{\frac{n-1}{n}}-1\right]+p_2'V_2'\left[\left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}}-1\right]\right\},$$

$p_1V_1=p_2'V_2'$, since h' is at the same temperature as a, hence

$$W=\frac{n}{n-1}p_1V_1\left[\left(\frac{p_2'}{p_1}\right)^{\frac{n-1}{n}}+\left(\frac{p_2}{p_2'}\right)^{\frac{n-1}{n}}-2\right].$$

This is a minimum when $\dfrac{dW}{dp_2'}=0$,

$$\frac{dW}{dp_2'}=\frac{n}{n-1}p_1V_1\left[\frac{n-1}{n}\left(\frac{p_2'}{p_1}\right)^{-\frac{1}{n}}\frac{1}{p_1}+\frac{n-1}{n}\left(\frac{p_2}{p_2'}\right)^{-\frac{1}{n}}\left(-\frac{p_2}{p_2'^2}\right)\right]=0,$$

$$\left(\frac{p_2'}{p_1}\right)^{-\frac{1}{n}}\left(\frac{p_2'^2}{p_1p_2}\right)=\left(\frac{p_2}{p_2'}\right)^{-\frac{1}{n}},$$

$$(p_2'^2)^{1-\frac{1}{n}}=(p_1p_2)^{1-\frac{1}{n}},$$

or

$$p_2'=\sqrt{p_1p_2}$$

is the condition for a minimum amount of work. Then

$$W=\frac{2n}{n-1}p_1V_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{2n}}-1\right].$$

For a three-stage compressor the condition for a minimum is

$$p_2' = \sqrt[3]{p_1^2 p_2},$$

$$p_2'' = \sqrt[3]{p_1 p_2^2},$$

or in general for m stages

$$W = \frac{mn}{n-1} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{mn}} - 1 \right],$$

$$p_2' = \sqrt[m]{p_1^{m-1} p_2},$$

$$p_2'' = \sqrt[m]{p_1^{m-2} p_2^2}, \quad p_2''' = \sqrt[m]{p^{m-3} p_2^3}, \quad \text{etc.}$$

The Harris Pneumatic pump is shown in Fig. 402. In this case to start the pump, air is sucked out of one cylinder A until it is filled with water, while the pump line on the other side is filled with water and air compressed into the cylinder B until the water is driven from the chamber. At this time the turn-over valve or switch M is shifted so that the compressed air on one side may fill the pipe on the other side. That is, if the volumes of the tanks are each represented by V and the volume of the pipe lines by NV where N is usually a fraction, the air of volume $(1+N)V$ in the discharging side at a pressure P_0 is connected with a volume NV in which the pressure is called p_0 (when filled with water). The temper-

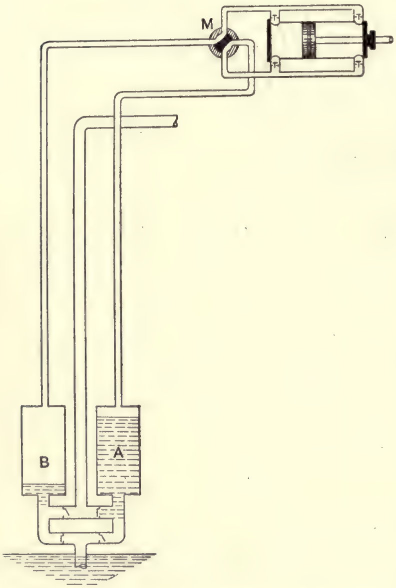


FIG. 402.—Harris Pneumatic Pump.

ature is constant in these pipes since they are exposed to the atmosphere and have much surface, hence when the valve M is switched

$$p_0NV+(1+N)VP_0=(1+2N)VP_1,$$

$$P_1=\frac{P_0(1+N)V+Np_0}{1+2N}.$$

After the pressure has fallen in the system to P_1 the compressor draws air from the tank just discharged and builds up the pressure in the pipe line leading to the full tank. After the pressure in that tank becomes equal to the discharge head P_0 the water begins to discharge. The foot valve on the empty tank does not open until sufficient air has been drawn out to bring the pressure to p_1. At that time water just begins to enter and when the tank is full, this pressure has become p_0. These two pressures, p_1 and p_0, do not differ by much since the head difference is the height of the tank, and this may be very small especially if the tank is on its side.

To find the quantity of air to be pumped from one tank to another there are three periods to consider.

1st. That in which the pressure in the pipe line leading to the full tank is brought from P_1 to P_0.

2d. That to reduce the pressure in the suction tank to p_1 while water is discharging from the full tank.

3rd. That to fill the suction tank.

To find the pressure in the suction tank at the end of the first period the following equation may be used:

$$P_0NV+P_{11}V(1+N)=P_1V(1+2N).$$

$$P_{11}=\frac{P_1(1+2N)-P_0N}{(1+N)}.$$

Since N is small, this quantity P_{11} will be found to be almost the same as P_1, but for exact work the formula must be used as stated and the displacement found.

Suppose the displacement of the compressor is D and this volume is abstracted from the volume $(1+N)V$ of the pipe and tank and discharged into the volume NV.

The ratio of $\frac{(1+N)V}{D}$ is M, and after the first stroke the pressure P_1 is reduced to $\frac{M}{M+1}P_1$, after the second to $\frac{M}{M+1}\left(\frac{M}{M+1}P_1\right)$ or $\left(\frac{M}{M+1}\right)^2 P_1$, and after the t strokes the pressure is $\left(\frac{M}{M+1}\right)^t P_1$.

This, however, equals P_{11}, hence

$$\left(\frac{M}{M+1}\right)^t P_1 = P_{11},$$

or

$$t = \frac{\log P_{11} - \log P_1}{\log M - \log(M+1)},$$

$tD = Q$, the quantity taken through the compressor during this first stage. During the second period the compressor compresses against a fixed discharge pressure but the suction is variable.

In this case

$$P_0 dV_y = P_x dV_c,$$

where dV is the variation of volume in the delivery tank, dV_c that in the compressor, and P_x the pressure in the tank being emptied. Now at any time

$$P_x V(1+N) + P_0(V_y + NV) = P_0 V(1+N) + p_0 NV.$$

Hence

$$\frac{P_0}{P_x} = \frac{V(1+N)}{V(1+N) + \frac{1}{r_0}NV - V_y - NV} = \frac{V(1+N)}{V\left(1+\frac{N}{r_0}\right) - V_y},$$

calling

$$r_0 = \frac{P_0}{p_0}.$$

Then

$$dV_c = \frac{P_0}{P_x} dV_y,$$

$$Q_2 = \text{Volume displaced} = \int_0^{v_1} \frac{P_0}{P_x}\, dV_y = \int_0^{v_1} \frac{V\left(1+\frac{N}{r_0}\right)}{V\left(1+\frac{N}{r_0}\right) - V_y}\, dV_y$$

$$= V(1+N) \log \frac{V\left(1+\frac{N}{r_0}\right)}{V\left(1+\frac{N}{r_0}\right) - V_1}.$$

V_1 is the volume of the discharge tank when the water is just entering the other tank, or the pressure is p_1 in that tank, hence

$$P_0(V_1+NV)+p_1V(1+N)=p_0NV+P_0(1+N)V,$$

$$P_0V_1 = V(P_0+p_0N-p_1N-p_1)$$
$$= V_0[P_0-p_1+N(p_0-p_1)].$$

Now N is a fraction and p_0-p_1 is very small, hence

$$V_1 = \frac{V}{P_0}(P_0-p_1).$$

Hence

$$Q_2 = V(1+N) \log \frac{V\left(1+\frac{N}{r_0}\right)}{V\left(1+\frac{N}{r_0}\right) - V\frac{P_0-p_1}{P_0}}$$

$$= V(1+N) \log \frac{1}{1-\frac{P_0-p_1}{P_0}\frac{r_0}{N+r_0}}$$

$$= V(1+N) \log \frac{1}{1-\frac{P_0-p_1}{P_0+Np_0}}.$$

Neglecting Np_0 in comparison with P_0

$$Q_2 = V(1+N) \log \frac{P_0}{p_1}.$$

Now, during the third period, or that of the filling of the suction chamber, the pressure in this chamber changes from

p_1 to p_0 when it is being filled with water, and since this change is so small the pressure may be assumed constant and the quantity removed during this time will be V.

The total displacement of the pump will then be

$$Q_1+Q_2+Q_3=Q,$$

$$tD+V(1+N)\log\frac{P_0}{p_1}+V=Q.$$

In this, V is the volume of the water lifted by the displacement of Q cubic feet of air by the compressor piston.

If the displacement of the compressor per revolution is D and it makes N revolutions per minute the time taken to fill one tank is:

$$\frac{Q}{ND}=t \text{ minutes.}$$

The effect of friction of the pipes, according to Harris, is to increase the pressure required by the amount.

$$\text{Loss}=KP=\frac{0.000,002}{14.7}\frac{l}{d}v^2P \text{ in pounds per sq.ft.}$$

l = length of pipe in feet;
d = diameter in inches;
v = velocity in ft. per sec.;
P = pressure in pounds per sq.ft.

The effect of this is to multiply variable pressures of discharge by $1+K$, or what is the same thing, to multiply the volume which passes through the compressor by $1+K$, and therefore the time is increased in the same manner.

The quantity K is seen to vary as the velocity of the air in the pipe varies, but for general use an average value of 0.1 is recommended by Harris. The quantity of air passing per second as the pressure changes is a variable, but the velocity of this air from the suction tank remains about the same because its pressure changes; while that entering the discharge tank varies with the weight, since the pressure is constant.

The size of the compressor can be determined when the

quantity of water to be lifted per hour is known, or the equation for Q may be used to get the quantity of water which may be handled by a given compressor.

The work done in the compressor when V_x cubic feet are compressed has been shown to be

$$W=\frac{np_xV_x}{n-1}\left[\left(\frac{P_0}{p_x}\right)^{\frac{n-1}{n}}-1\right],$$

where n is the exponent of the compression curve. This quantity W changes in value as the air is drawn out. To find when it is a maximum express it in terms of the variable p_x, as V_x is constant, depending on the displacement of the compressor while P_0 is a constant,

$$\text{work}=\frac{nV}{n-1}\left[p_x^{\frac{1}{n}}P_0^{\frac{n-1}{n}}-p_x\right].$$

This is a maximum when

$$\frac{dW}{dp_x}=0=\frac{nV}{n-1}\left[\frac{1}{n}p_x^{\frac{1-n}{n}}P_0^{\frac{n-1}{n}}-1\right],$$

$$p_x=\left(\frac{1}{n}\right)^{\frac{n}{n-1}}P_0.$$

For $n=1.4$; $p_x=0.309P_0$.

Putting this in for p_x the work for V cubic feet passing through the compressor becomes:

$$W=\frac{nV}{n-1}\left[\left(\frac{1}{n}\right)^{\frac{1}{n-1}}P_0-\left(\frac{1}{n}\right)^{\frac{n}{n-1}}P_0\right]$$

$$=\left(\frac{1}{n}\right)^{\frac{1}{n-1}}\frac{nP_0V_1}{n-1}\left[1-\left(\frac{1}{n}\right)\right]=\left(\frac{1}{n}\right)^{\frac{1}{n-1}}P_0V_1.$$

If the displacement of the compressor is ND where D is the discharge per revolution, the horse-power which must be cared for is:

$$\text{Max. H.P.}=\left(\frac{1}{n}\right)^{\frac{1}{n-1}}\frac{P_0ND}{33{,}000}.$$

This is divided by the efficiency of the compressor and driving motor to get the applied power at this instant.

To get the total power employed, the following method will be used, for the second stage:

$$dW = \left[(P_0)^{\frac{n-1}{n}} P_x^{\frac{1}{n}} - P_x\right]\frac{ndV}{n-1}$$

$$W = \frac{nP_0^{\frac{n-1}{n}}}{n-1}\int_{p_1}^{P_{\prime\prime}} P_x^{\frac{1}{n}} dV - \frac{n1}{n-1}\int_{p_1}^{P_{\prime\prime}} P_x dV.$$

Now although the compression within the cylinder is according to the law $pV^n = K$ the relation between the pressures and volumes in the tanks and pipe lines is isothermal, so that at any time

$$p_x(1+N)V + P_0(V_x+NV)p_x' = (1+N)V + P_0(V_2+NV),$$

or considering a small amount, dV taken from the volume $V(1+N)$ the change becomes

$$P_x dV = V(1+N)dP_x,$$

then
$$dV = \frac{V(1+n)}{P_x}dP_x;$$

hence

$$W_2 = n\frac{V(1+N)}{n-1}P_0^{\frac{n-1}{n}}\int P_x^{\frac{1-n}{n}} dP_x - \frac{V(1+N)}{n-1}n\int dP_x$$

$$= \frac{n^2}{n-1}V(1+N)P_0^{\frac{n-1}{n}}\left[P_{\prime\prime}^{\frac{1}{n}} - p_1^{\frac{1}{n}}\right] - \frac{nV(1+N)}{n-1}(P_{\prime\prime} - p_{.})$$

$$= \frac{n(1+N)}{n-1}V\left[nP_0^{\frac{n-1}{n}}\left(P_{\prime\prime}^{\frac{1}{n}} - p_1^{\frac{1}{n}}\right) - (P_{\prime\prime} - p_1)\right].$$

The work done during the first period is

$$W_1 = \frac{n'D}{n-1}\left[\left(\frac{P_0+P_1}{2}\right)^{\frac{n-1}{n}} P_1^{\frac{1}{n}} - P_1\right],$$

and the work on the last period is

$$W_3 = \frac{nV}{n-1}\left[P_0^{\frac{n-1}{n}} p_1^{\frac{1}{n}} - p_1\right].$$

The useful work is

$$V(P_0-p_1)$$

Hence the efficiency of this method of pumping is

$$\text{Eff.}=\frac{V(P_0-p_1)}{(W_1+W_2+W_3)}\times \text{eff. compressor and motor.}$$

Harris has designed for this method of pumping a special switch which will reverse the turnover valve after the compressor has made the necessary number of turns tN, or by a system of diaphragms the valve is changed when the pressure p_0 is reached. Leakage from the system is made up from the atmosphere and allows a supply of air when the pressure in the suction is below the atmosphere before the valve reverses. If an excess of air is drawn in this will do no harm, as it will be blown out of the discharge before the valve reverses the action.

It is recommended by the builders of this pump to make the air pipe of such a diameter that the maximum velocity is not over 5000 feet per minute when it is assumed to have the volume of the free air. That is

$$\text{Area air pipe}=\frac{VP_0}{t\times 14.7\times 144\times 5000}$$

The tanks are usually made of such a volume that $N=\frac{1}{4}$, or tanks are four times the volumes of the

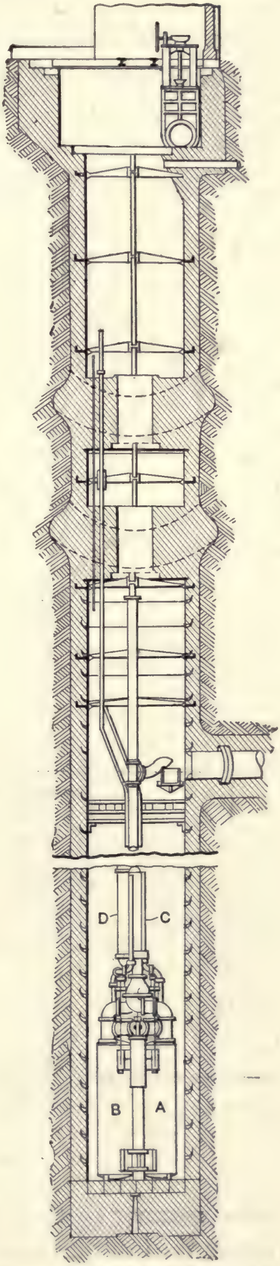


FIG. 403.—Harris Pneumatic Pump.

pipes. The discharge pipes for the water are made 2.8 diameter of air pipes and they should discharge into a head in the

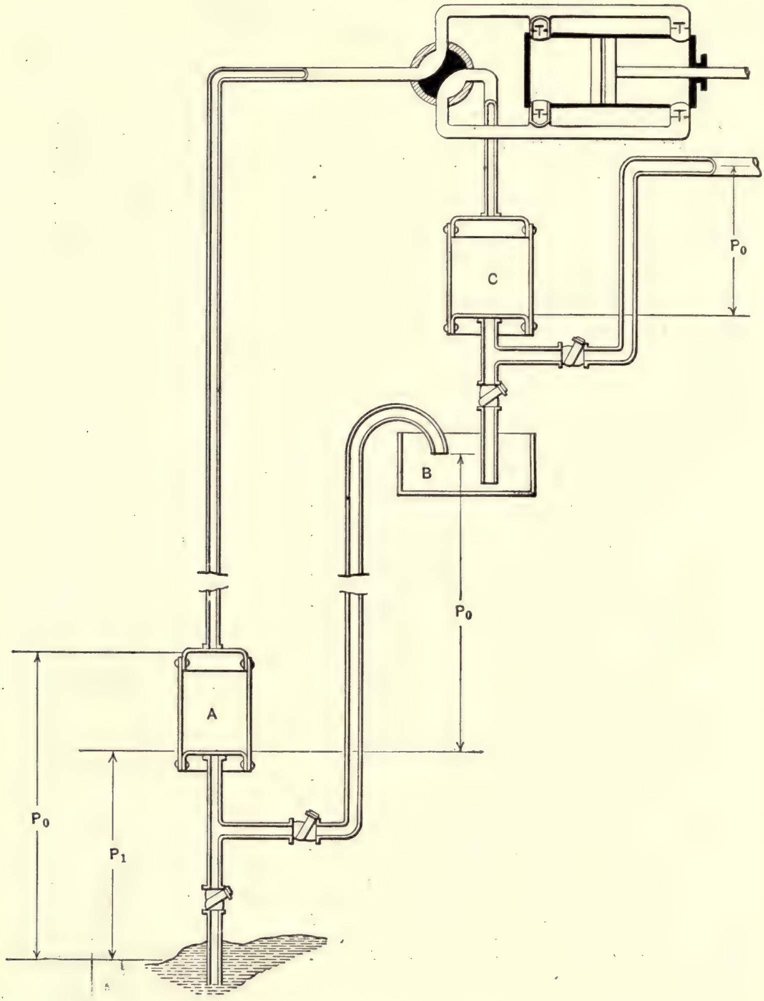


FIG. 404.—Separate Stage Pump.

same manner as that shown for the air lift pumps. Fig. 403 shows the installation of a Harris pneumatic pump for draining the inverted syphon at High Bridge. The tanks *AB* are con-

nected to the compressor by the pipes *CD* which extend to the power-house.

Fig. 404 illustrates a method of using the system for high lifts where it is desired to use an air pressure less than that corresponding to the total lift. The air forced into *A* lifts the water against the pressure P_0 into the suction box *B* while at this time the tank *C* is being filled. When the pump reverses the water in *C* is discharged while *A* is filled by suction.

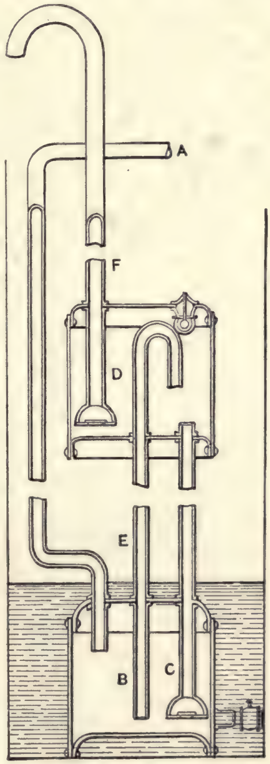


FIG. 405.—Differential Air Lift Pump.

The method shown in Fig. 405 is intended to use compressed air under a moderate pressure to raise water a considerable height. Air is delivered into *B* through *A* at sufficient pressure to force water through the foot valve *C* into the tank *D*. When the air which rises to the top of *B* has driven the water from *B* so that the bottom of the pipe *E* is exposed, the air supply is cut off and the air in *B* escapes and passes up to the next tank and lifts the water from it through the foot valve in *F*. The air which filled tank *B* will fill tanks *B* and *D* finally and hence the pressure will be half the pressure of the supply if the two tanks are of equal size. This means the second lift can only be made half of the lift between the first and second tanks. If a third tank is used the next lift will be one third the original lift.

In this method the air is finally discharged and the only use of the internal energy of the air is in the successive lifts.

After the air has been driven out from the upper tank the

lower tank is allowed to fill as it is open to the atmosphere through E and D and water will flow in by gravity.

There are other compressed air pumps which act the same as these, but the principles here brought out should serve to make their action clear.

CHAPTER XIV

CENTRIFUGAL PUMPS

THE last few years have seen many improvements in centrifugal pumps; much higher efficiencies have been obtained and greater lifts have been overcome. These gains have resulted

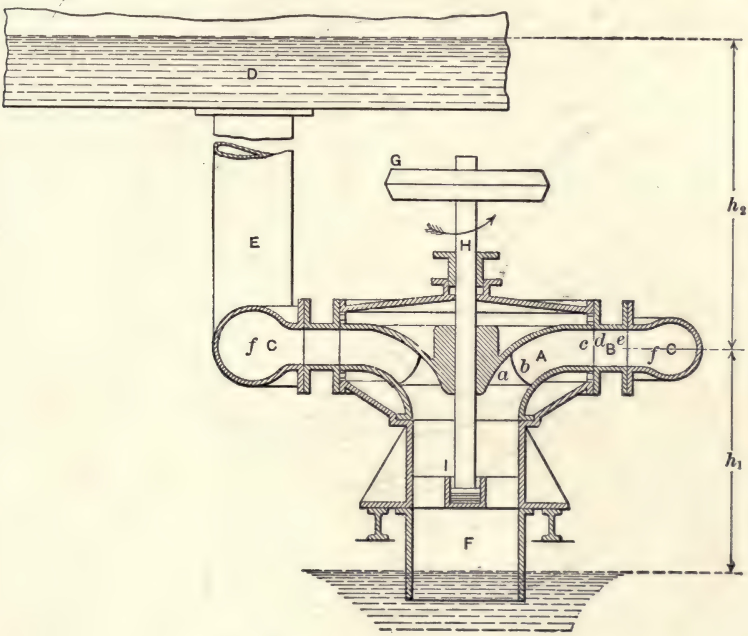


FIG. 406.—Suction of Centrifugal Pump.

from a careful study of the principles of the pump and from the theoretical design of pumps rather than from the empirical design which was customary in earlier times. To understand the theory of these pumps an examination will be made of Figs. 406 and 407. Fig. 406 is a longitudinal section through

a single-stage pump, while Fig. 407 is a cross-section through the *impeller* or *runner* A and the *diffuser vanes* B. With water in the discharge tank D at the level shown in the figure, the whole system will be filled with water and this water will tend to flow from the tank through the pipe E, the *volute casing* C,

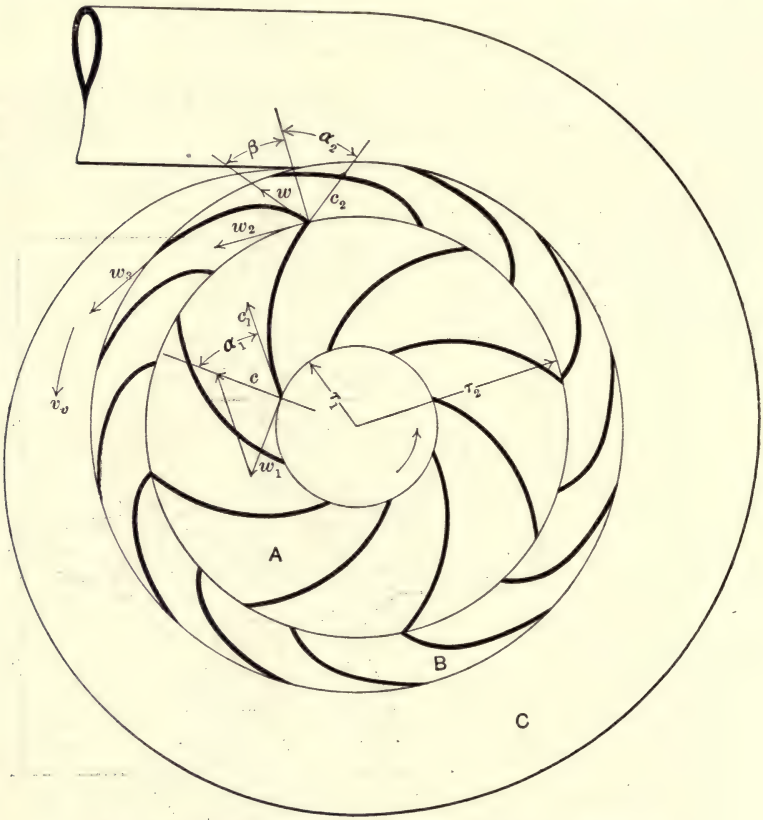


FIG. 407.—Section through Impeller, Diffuser and Casing.

the *diffuser* B, the *runner* A and the *suction pipe* F. The runner A is mounted on the shaft H which is supported by the *foot bearing* I. If the shaft is horizontal two bearings take the load resulting from the weight of the parts.

If now the shaft H be driven by a belt on the pulley G or by a motor directly connected with the shaft, the water is

driven by the runner and forced to move outward by the shape of the vanes and by centrifugal force; moreover this motion takes place in such a manner that the water leaving the outer tips of the runner enters the vanes in the diffuser with no shock. In the diffuser the direction of motion is changed to one more nearly the direction to be obtained in the volute chamber and also the velocity is decreased so that velocity head is changed into pressure head. If the speed of the runner is sufficiently great the water will acquire enough velocity that when the ends of the vanes B are reached the pressure will be greater than that due to the head of the tank D.

As the water is driven outward on the wheel a partial

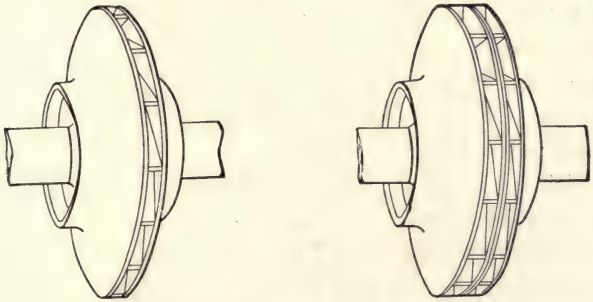


FIG. 408.—Single and Double-flow Impellers.

vacuum is produced at the center and water is drawn up to fill the space. At times this water is drawn in on two sides of a disc, giving a **double-flow pump.** The shaft is often placed in a horizontal position, but for simplicity in the derivations of equations the shaft has been shown here in a vertical position. In some cases the vanes of the diffuser are omitted and in this event the space around the wheel is sometimes called the *whirlpool chamber*. There may be guide vanes placed in the suction at the point of entrance to the runner for the purpose of making the direction of the water entering the runner more definite. Fig. 408 illustrates the two forms of runner.

A study will now be made of the pressures in the pump

and of the action of the pump in which the following symbols will be used:

r_1 = radius of point of entrance $= \dfrac{D_1}{2}$;

r_2 = radius of point of exit $= \dfrac{D_2}{2}$;

ζ = coefficient of lost head (see Chapter V);
c = absolute velocity of water at entrance;
u = absolute velocity of water at exit from impeller or entrance to diffuser;
c_1 = velocity at entrance relative to impeller;
c_2 = velocity at exit relative to impeller;
v_s = velocity in suction pipe;
v_d = velocity in discharge pipe;
v_v = velocity in volute;
u_3 = velocity at exit from diffuser;
v_0 = velocity in outlet or discharge pipe;
A = area perpendicular to c;
A with subscript = area perpendicular to velocity with same subscript;
w_1 = velocity of wheel or impeller at entrance;
w_2 = velocity of impeller at exit;
ω = angular velocity;
N = R.P.M.;
h_1 = suction head to center of wheel;
h_2 = discharge head from center of wheel;
$H = h_1 + h_2$ = total net head;
p_a, p_b, p_c, p_d, etc., pressure at various points in feet head;
w = weight of 1 cu. ft. of water;
a = pressure of the atmosphere in feet;
g = acceleration of gravity;

All the above are expressed in feet and pounds.

At the point of entrance into runner, the pressure in the casing p_a is given by the equation,

$$p_a = a - h_1 - \zeta_s \frac{v_s^2}{2g} - \frac{c^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

When the water enters the runner, the portion of p_a which remains to overcome the friction of the passage of the runner and to accelerate the water in so far as the section of this passage may change is called p_b. This is given by:

$$p_b = p_a - \frac{w_1^2}{2g} - \zeta_{ent}\frac{c^2}{2g} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The term $\frac{w_1^2}{2g}$ is the pressure head due to the action of centrifugal force.* This action, which exists in all rotating bodies containing fluids, is to produce a hydrostatic pressure equal to the above expression. This, like any other hydrostatic pressure, acts in all directions. Where there is a free surface as in a cup, this manifests itself by causing the water to pile up where the velocity is the greatest in such a manner that the free surface is a paraboloid of revolution. Where there is no free surface, there is a force equal to the height to which the water would rise above the point considered were a free surface possible. This force is equal to $\frac{w_x^2}{2g}$. The actual force on a rotating body which would cause flow and overcome friction

---

* In Fig. 409, the particle of water in the free surface of the rotating vessel at a distance x from the center is moving with a velocity $x\omega$, if ω is the angular velocity. It is subject to the centrifugal force $\frac{w}{g}\omega^2 x$ expressed in pounds, while the force w acts vertically, due to gravity. The resultant of these two forces acts normal to the free surface, hence

$$\frac{dy}{dx} = \frac{C.F}{w} = \frac{w\omega^2 x}{gw} = \frac{\omega^2 x}{g},$$

$$y = \int_0^x \frac{\omega^2 x}{g} dx = \frac{\omega^2 x^2}{2g} = \frac{v_x^2}{2g}.$$

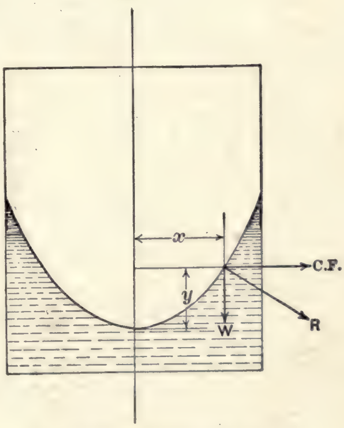


FIG. 409.—Revolving Vessel.

is not the total force acting at a point, but that force minus $\frac{w_x^2}{2g}$.

$$\text{At } c, \quad p_c = p_b + \frac{c_1^2}{2g} - \frac{\zeta_v c_1^2}{2g} - \frac{c_2^2}{2g}. \quad \ldots \quad (3)$$

$$\text{At } d, \quad p_d = p_c + \frac{w_2^2}{2g} - \zeta_{ex}\frac{u^2}{2g}. \quad \ldots \quad (4)$$

$$\text{At } e, \quad p_e = p_d + \frac{u^2}{2g} - \zeta_d\frac{u^2}{2g} - \frac{u_3^2}{2g}. \quad \ldots \quad (5)$$

If $u_3 = v_v$ there is no loss at discharge into the volute, but if this is not the case, then there is a sudden change in velocity and there is a loss

$$\frac{(u_3 - v_v)^2}{2g} = \text{loss}, \quad \ldots \quad (6)$$

and

$$p_v = p_e + \frac{u_3^2}{2g} - \frac{(u_3 - v_v)^2}{2g} - \frac{v_v^2}{2g}, \quad \ldots \quad (7)$$

but

$$p_v + \frac{v_v^2}{2g} = h_2 + a + \zeta_0\frac{v_0^2}{2g} + \frac{v_0^2}{2g}. \quad \ldots \quad (8)$$

Substituting for p_v, p_e, p_d, p_c, p_b, etc., their values from the preceding equations the following results after collecting and rearranging:

$$h_2 + h_1 + \zeta_0\frac{v_0^2}{2g} + \frac{(u_3 - v_v)^2}{2g} + \zeta_d\frac{u^2}{2g} + \zeta_{ex}\frac{u^2}{2g} + \zeta_v\frac{c_1^2}{2g} + \zeta_{ent}\frac{c^2}{2g}$$

$$+ \zeta_s\frac{v_s^2}{2g} + \frac{v_0^2}{2g} = \frac{u^2}{2g} + \frac{w_2^2}{2g} + \frac{c_1^2}{2g} - \frac{c_2^2}{2g} - \frac{w_1^2}{2g} - \frac{c^2}{2g}. \quad (9)$$

The passages of the pump are completely filled with water, hence

$$Ac = A_1c_1 = A_2c_2 = A_{ex}u = A_3u_3, \text{ etc.} \quad \ldots \quad (10)$$

From these equations it is seen that all of the velocities may

be expressed in terms of the areas and any other velocity. For example:

$$c=\frac{A_{\text{ex}}}{A}u, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$\therefore \quad \zeta_{\text{ent}}\frac{c^2}{2g}=\zeta_{\text{ent}}\left(\frac{A_{\text{ex}}}{A}\right)^2\frac{u^2}{2g}=\zeta'_{\text{ent}}\frac{u^2}{2g}. \quad . \quad . \quad . \quad (12)$$

The Eq. (9) now becomes

$$h_2+h_1+(\zeta_0'+\zeta'_{s.e.}+\zeta_d+\zeta'_{\text{ex}}+\zeta'_v+\zeta'_{\text{ent}}+\zeta'_s)\frac{u^2}{2g}+\frac{v_0^2}{2g}$$
$$=\frac{1}{2g}[u^2+w_2^2-c_2^2-(c^2+w_1^2-c_1^2)]. \quad (13)$$

If there is to be no impact at entrance and exit, the velocities

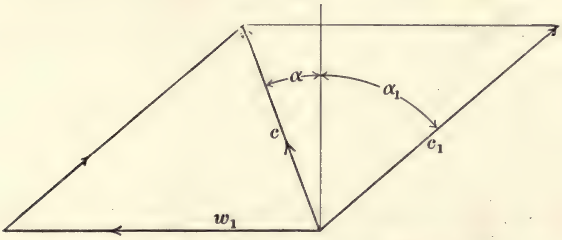


FIG. 410.—Velocities at Entrance—α positive, α_1 negative.

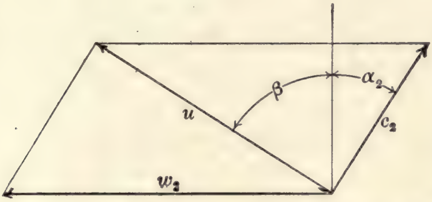


FIG. 411.—Velocities at Discharge—α_2 negative, β positive.

at those points must have certain relations which are seen from the parallelogram of velocities. Calling the angle between the radius and c, α; between the radius and c_1, α_1; between the radius and c_2, α_2; and finally that between u and the radius, β; the following relations may be seen from Figs. 407, 410 and 411. (The angles are taken as positive when they are measured on the side of the radius toward which the motion w_1 is taking

place, provided the lines be drawn away from the point considered and in direction of the velocity.)

$$c_1 \cos \alpha_1 = c \cos \alpha \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

$$c \sin \alpha - c_1 \sin \alpha_1 = w_1; \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

$$c_1^2 = c^2 + w_1^2 - 2cw_1 \sin \alpha \text{ or } c^2 + w_1^2 - c_1^2 = 2cw_1 \sin \alpha; \quad (16)$$

$$c_2 \cos \alpha_2 = u \cos \beta; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$u \sin \beta - c_2 \sin \alpha_2 = w_2; \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$c_2^2 = u^2 + w_2^2 - 2uw_2 \sin \beta \text{ or } u^2 + w_2^2 - c_2^2 = 2uw_2 \sin \beta. \quad (19)$$

Substituting (16) and (18) in Eq. (13) there results:

$$H + (\Sigma\zeta)\frac{u^2}{2g} + \frac{v_0^2}{2g} = \frac{1}{g}[uw_2 \sin \beta - cw_1 \sin \alpha]. \quad . \quad . \quad (20)$$

Calling $\Sigma\zeta \frac{u^2}{2g}$ the lost head, lH, and $\frac{v_0^2}{2g}$ the residual velocity head, rH, the left-hand side of the equation represents the total head to be accounted for. This when multiplied by one pound is the number of foot-pounds of work done on each pound of water passing from the suction forebay to the discharge tank.

$$H(1+l+r) = \text{work per pound of water}$$

$$= \frac{1}{g}(uw_2 \sin \beta - cw_1 \sin \alpha).$$

To get the total work this expression must be multiplied by the quantity of water passing through the pump. All of the water taken into the pump at *a* does not leave the discharge point of the diffuser, for a certain amount leaks past the clearance space between the wheel and the diffuser or casing. The quantity leaking from this point will be computed, but for the

present it will be called Qe. The amount desired is Q cubic feet per second, and so the amount Qp, to be pumped is

$$Qp = Q + Qe; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (21)$$

$$QpwH(1 + l + r) = \text{Work per sec.}; \quad . \quad . \quad . \quad . \qquad (22)$$

$$Qpw\frac{1}{g}(uw_2 \sin \beta - cw_1 \sin \alpha) = \text{Work per sec.} \qquad (23)$$

The expressions above are those which give the work required to do the hydraulic work, but in addition to this there must be energy supplied to overcome the resistance of the friction of the bearings, stuffing boxes, and the leakage water against the runner. This latter quantity may be quite large, while the second one depends on many variables. The first may be made very small by proper lubrication. These will be discussed later, but for the present they may be written as $f_w W_u$, $f_b W_u$ and $f_s W_u$. The total work then becomes

$$\left.\begin{aligned} W &= W_u(1 + f_w + f_b + f_s) \\ &= \frac{wQ}{g}p(uw_2 \sin \beta - cw_1 \sin \alpha)(1 + f_w + f_b + f_s) \\ &= wQpH(1 + l + r)(1 + f_w + f_b + f_s) \\ &= w(Q + Q_e)H(1 + l + r)(1 + f_w + f_b + f_s) \end{aligned}\right\} \quad . \quad . \qquad (24)$$

The useful work is wQH or in some cases, $wQH(1 + r)$, if the residual velocity may be utilized in any manner. The efficiency then becomes

$$Eff = \frac{wQH}{W} = \frac{1}{(1 + e)(1 + l + r)(1 + f_w + f_b + f_s)}. \quad . \qquad (25)$$

The efficiency is seen to vary with the values of these coefficients; they should all be as small as possible. The determination of these quantities will be considered after a discussion of the action of the water in the wheel.

There are several relations which may be determined from

Figs. 407, 410 and 411. Fig. 412 gives the two diagrams of Figs. 409 and 410 in one.

α is often made 0° so that the work equation becomes

$$H(1+l+r)=\frac{w_2 u \sin \beta}{g}=kH. \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$k=1+l+r;$$

now
$$w_2=w_1\frac{r_2}{r_1};$$

$$w_1=c_1 \sin \alpha_1;$$

$$c_1=\frac{A_{ex}u}{A_1}.$$

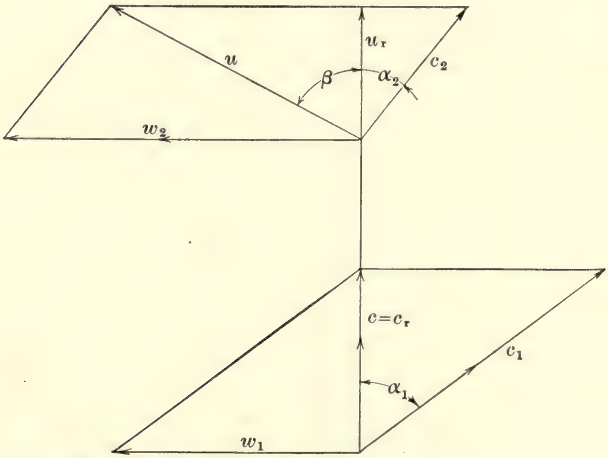


FIG. 412.—Diagrams at Entrance and Exit.

Hence by combining these

$$kH=\frac{r_2}{r_1}\frac{A_{ex}}{A_1}\frac{u^2}{g}\sin \beta \sin \alpha_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

or
$$u=\sqrt{\frac{2g(kH)}{2\frac{r_2}{r_1}\frac{A_{ex}}{A_1}\sin \beta \sin \alpha_1}}. \quad . \quad . \quad . \quad . \quad . \quad (28)$$

If it is desired to express this in terms of the effective head H, Eq. (20) may be written:

$$H=\frac{u^2}{2g}\left[2\frac{r_2}{r_1}\frac{A_{ex}}{A_1}\sin\beta\sin\alpha_1-\Sigma\zeta-\left(\frac{A_{ex}}{A_d}\right)^2\right], \quad . \quad . \quad . \quad (29)$$

$$u=\sqrt{\frac{2gH}{2\frac{r_2}{r_1}\frac{A_{ex}}{A_1}\sin\beta\sin\alpha_1-\Sigma\zeta-\left(\frac{A_{ex}}{A_d}\right)^2}}. \quad . \quad . \quad . \quad (30)$$

If either of these equations be used, the losses must be expressed in terms of H or u before u can be found. The ratios $\frac{A_{ex}}{A_1}$ and $\frac{r_2}{r_1}$ may be assumed from practice as well as the ratio $\frac{A_{ex}}{A_d}$.

For convenience it is well to take the components of u and c in a radial direction; these will be called u_r and c_r.

$$u_r=u\cos\beta=c_2\cos\alpha_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad (31)$$

$$c_r=c\cos\alpha=c_1\cos\alpha_1. \quad . \quad . \quad . \quad . \quad . \quad (32)$$

Neumann uses the total head kH in finding the velocities u_1, u_r and c_2 in the following manner:

$$\frac{1}{g}uw_2\sin\beta=kH, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

$$\sin\beta=\frac{gkH}{uw_2}.$$

From Fig. 412, Eq. (18) and Eq. (31),

$$w_2=u\sin\beta-c_2\sin\alpha_2=\frac{ugkH}{uw_2}-u_r\tan\alpha_2;$$

$$w_2=\frac{gkH}{w_2}-u_r\tan\alpha_2, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

$$w_2=-\tfrac{1}{2}u_r\tan\alpha_2+\sqrt{gkH+\left(\frac{u_r\tan\alpha_2}{2}\right)^2}. \quad . \quad . \quad (34)$$

For $\alpha_2 = 0°$ this becomes

$$w_{2,\,0°} = \sqrt{gkH}.$$

Since $$u_r = u\cos\beta = u\sin\beta\cot\beta = \frac{gkH\cot\beta}{w_2}, \quad . \quad . \quad . \quad (34a)$$

Eq. (33) may be written:

$$w_2 = \frac{gkH}{w_2} - \frac{gkH}{w_2}\tan\alpha_2\cot\beta, \quad . \quad . \quad . \quad . \quad (35)$$

$$w_2 = \sqrt{gkH\left(1 - \frac{\tan\alpha_2}{\tan\beta}\right)} = x\sqrt{gkH}, \quad . \quad . \quad (36)$$

calling $$x = \sqrt{\left(1 - \frac{\tan\alpha_2}{\tan\beta}\right)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Eq. (33) may be used to determine the angle α_2 in case w_2, and u_r are known, and Eq. (35) may be used if w_2 and β are known. Thus

$$\tan\alpha_2 = \frac{\frac{gkH}{w_2} - w_2}{u_r}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

$$\tan\alpha_2 = \frac{\frac{gkH}{w_2} - w_2}{\frac{gkH}{w_2}\cot\beta} = \frac{1 - \frac{w_2^2}{gkH}}{\cot\beta}. \quad . \quad . \quad . \quad . \quad . \quad (39)$$

For the relative velocity c_2, Neumann uses the equations,

$$c_2 = \frac{u_r}{\cos\alpha_2},$$

or $$c_2 = \frac{u\cos\beta}{\cos\alpha_2} = \frac{u\sin\beta}{\tan\beta\cos\alpha_2} = \frac{gkH}{w_2\tan\beta\cos\alpha_2}. \quad . \quad (40)$$

For u:

$$u = \frac{gkH}{w_2\sin\beta},$$

$$u = \frac{gkH}{\sin\beta\sqrt{gkH\left(1 - \frac{\tan\alpha_2}{\tan\beta}\right)}} = \sqrt{\frac{gkH}{1 - \frac{\tan\alpha_2}{\tan\beta}}(1+\cot^2\beta)}. \quad (41)$$

These equations are used when certain quantities are known or assumed. The assumptions fix the equations which would be used. These quantities are also obtained in another manner by Neumann. He uses the expression

$$w_2 = x\sqrt{gkH}$$

and for u_r he proceeds as follows:

$$u_r = \frac{gkH \cot \beta}{w_2}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

when $\alpha_2 = 0°$, $\qquad w_2 = \sqrt{gkH}$. . . . . (43)

Hence $\qquad u_{r0°} = \sqrt{gkH} \cot \beta_0° = c_2 \cos 0° = c_2$, ... (44)

calling $\cot \beta = \sqrt{2\lambda}$,

$$u_{r0°} = \sqrt{\lambda 2gkH}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

Using the value of $w_2 = x\sqrt{gkH}$,

$$u_r = \frac{gkH \cot \beta}{x\sqrt{gkH}} = \frac{\sqrt{gkH} \cot \beta}{x}, \quad . \quad . \quad . \quad . \quad (46)$$

and if in general $\cot \beta = \sqrt{2\lambda}$ or $\lambda = \frac{1}{2} \cot^2 \beta$, the result follows:

$$u_r = \frac{\sqrt{\lambda 2gkH}}{x}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

from Eq. (41),

$$u = \frac{\sqrt{gkH}}{x}\sqrt{1 + 2\lambda}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

The author has re-computed and constructs the curves shown in Figs. 413 and 414, which are those used by Neumann. By assuming various angles for α_2 and β the values of u, w_2, and u_r may be found in terms of kH.

A series of curves drawn by the use of the equation,

$$w_2 = x\sqrt{gkH}$$

shows how w_2 varies with different values of β and α_2, but

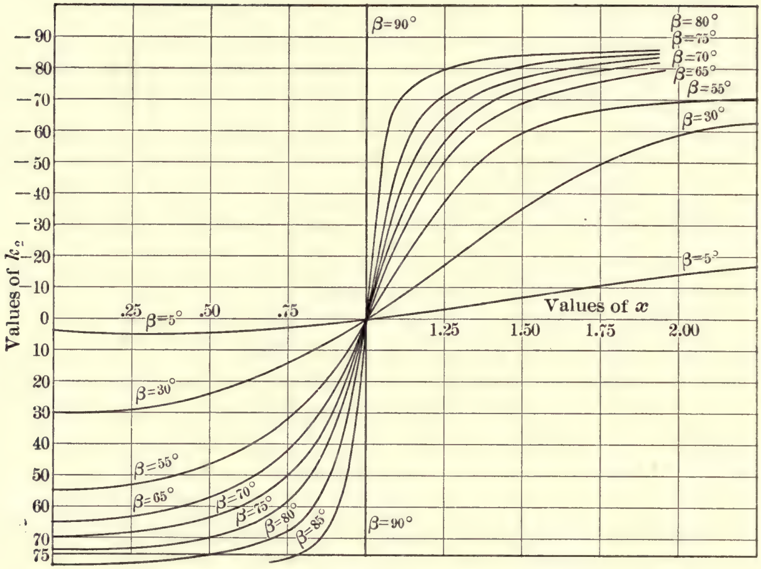


FIG. 413.—Values of x. (After Neumann.)

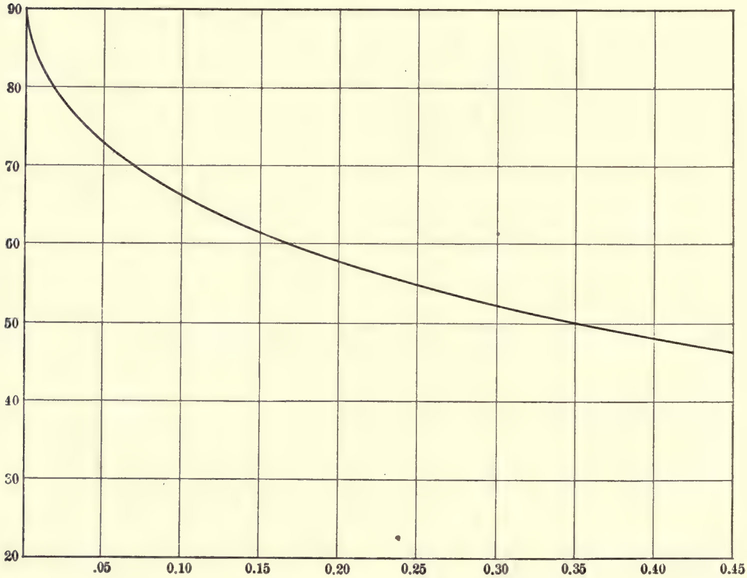


FIG. 414.—Values of λ. (After Neumann.)

since $w_2 = x$ when $\sqrt{gkH} = 1$, these curves need not be drawn, as the x curve shows this variation to some scale.

It is seen from an examination of Fig. 413 that when $\beta = 90°$, $x = 1$ for all values of α_2 except $90°$ and that for $\alpha_2 = 0$,

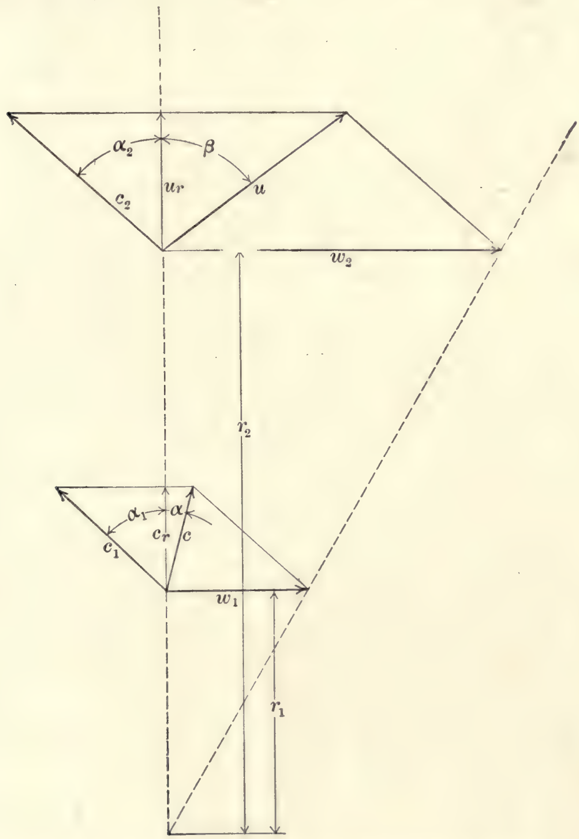


FIG. 415.—Combined Entrance and Discharge Diagram.

all values of x for different values of β are unity. The variation in x and consequently w_2 is greater for changes in α_2 for smaller values of β than for values of β near $90°$. When β has a value of $85°$ there is little change, as α_2 changes from $50°$ to $-50°$. When considerable variation is desired for different values of α_2, β must be taken smaller, say $60°$.

When α is not 0° the equations just found will be of different forms. The entrance velocities will now be considered. A new figure, using the construction for $w_2 = \frac{w_1}{r_1} r_2$, is given in Fig. 415, showing these other relations. In this figure, $c_1 = c_2 \frac{A_2}{A_1}$. Now $c_r = c \cos \alpha = c_1 \cos \alpha_1$,

$$c \sin \alpha = c_r \tan \alpha. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

$$u \sin \beta = w_2 + c_2 \sin \alpha_2 = w_2 + u_r \tan \alpha_2. \quad . \quad . \quad (50)$$

Now
$$\frac{1}{g}(uw_2 \sin \beta - cw_1 \sin \alpha) = kH,$$

or
$$w_2^2 + w_2 u_r \tan \alpha_2 - w_1 c_r \tan \alpha = gkH. \quad . \quad . \quad . \quad (51)$$

$$w_1 = w_2 \frac{r_1}{r_2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

Hence

$$w_2 = \frac{r_1}{r_2} \frac{c_r \tan \alpha}{2} - \frac{u_r}{2} \tan \alpha_2 + \sqrt{\left[\frac{r_1}{r_2} \frac{c_r \tan \alpha}{2} - \frac{u_r \tan \alpha_2}{2}\right]^2 + gkH.}$$

Now $u_r 2\pi r_2 b_2 = c_r 2\pi r_1 b_1$, hence

$$w_2 = \frac{c_r}{2}\left[\frac{r_1}{r_2} \tan \alpha - \frac{r_1 b_1}{r_2 b_2} \tan \alpha_2\right]$$

$$+ \sqrt{\left[\frac{r_1}{r_2} \tan \alpha - \frac{r_1 b_1}{r_2 b_2} \tan \alpha_2\right]^2 \frac{c_r^2}{4} + gkH}. \quad (53)$$

In one of these, c_r must be assumed as well as $\frac{r_1}{r_2}$, $\frac{b_1}{b_2}$, α_2, and α, while in the other c_r, u_r, $\frac{r_1}{r_2}$, α_2, and α are assumed.

If the velocity u is desired in terms of the effective head H, the following results:

$$H=\frac{1}{g}(uw_2\sin\beta-cw_1\sin\alpha)-\Sigma\zeta\frac{u^2}{2g}-\left(\frac{A_{ex}}{A_d}\right)^2\frac{u^2}{2g};$$

$$w_2=u\sin\beta-c_2\sin\alpha_2;$$

$$c_2\cos\alpha_2=u\cos\beta;$$

$$w_2=u\sin\beta-u\cos\beta\tan\alpha_2=u(\sin\beta-\cos\beta\tan\alpha_2); \quad . \quad (54)$$

$$w_1=c\sin\alpha-c_1\sin\alpha_1=c(\sin\alpha-\cos\alpha\tan\alpha);$$

$$c=\frac{A_{ex}}{A}u;$$

$$2gH=2u^2\Bigg[\sin\beta^2-\cos\beta\tan\alpha_2\sin\beta$$

$$-\left(\frac{A_{ex}}{A}\right)^2[\sin\alpha-\cos\alpha\tan\alpha_1]\sin\alpha-\Sigma\zeta-\left(\frac{A_{ex}}{A_d}\right)^2\Bigg];$$

$$u=\sqrt{\frac{2gH}{2[\sin\beta-\cos\beta\tan\alpha_2]\sin\beta-\left(\frac{A_{ex}}{A}\right)^2[\sin\alpha-\cos\alpha\tan\alpha_1]\sin\alpha-\Sigma\zeta-\left(\frac{A_{ex}}{A_d}\right)^2}} \quad (55)$$

and substituting the value of u above, w_2 is found. Another method gives the following:

$$w_2u\sin\beta-w_1c\sin\alpha=gkH;$$

$$u\sin\beta=\frac{gkH}{w_2}+\frac{r_1}{r_2}c_r\tan\alpha;$$

$$u=\frac{u_r}{\cos\beta},$$

$$u_r=\frac{1}{\tan\beta}\left[\frac{gkH}{w_2}+\frac{r_1}{r_2}c_r\tan\alpha\right]. \quad . \quad . \quad . \quad . \quad (56)$$

Now $$w_2=u\sin\beta-u_r\tan\alpha_2,$$

$$w_2=\frac{gkH}{w_2}+\frac{r_1}{r_2}c_r\tan\alpha-u_r\tan\alpha_2.$$

Substituting the value of u_r from Eq. (56),

$$w_2 = \frac{gkH}{w_2}\left[1 - \frac{\tan\alpha_2}{\tan\beta}\right] + \frac{r_1}{r_2} c_r \tan\alpha \left[1 - \frac{\tan\alpha_2}{\tan\beta}\right]. \qquad (57)$$

$$w_2 = \frac{r_1}{r_2} c_r \frac{\tan\alpha}{2}\left[1 - \frac{\tan\alpha_2}{\tan\beta}\right]$$

$$+ \sqrt{gkH\left[1 - \frac{\tan\alpha_2}{\tan\beta}\right] + \left(\frac{r_1}{r_2} c_r \frac{\tan\alpha}{2}\left[1 - \frac{\tan\alpha_2}{\tan\beta}\right]\right)^2}$$

$$= \frac{r_1}{r_2} c_r \frac{\tan\alpha}{2}(x) + \sqrt{gkHx + \left(\frac{r_1}{r_2} c_r \frac{\tan\alpha}{2}\right)^2 (x)^2}. \qquad (58)$$

These equations may be used in determining the velocities of various parts of the wheel for a given set of conditions.

Losses

The first loss to be considered in Eq. (9) is the loss in the suction pipe ζ_s. This coefficient includes the losses due to entrance, friction in pipe and bends. These losses have been discussed in Chapter V, and from what has been given there the following may be written:

$$\zeta_s \frac{v_s^2}{2g} = \left[\tfrac{1}{2} + f\frac{l}{d} + m\right]\frac{v_s^2}{2g}. \qquad (59)$$

The equation shows that ζ_s varies directly with l and inversely with d, and moreover the total loss varies with v_s^2. It is an important matter to determine just how much $\zeta_s \frac{v_s^2}{2g}$ may be. In any case

$$p_a = a - h_1 - \zeta_s \frac{v_s^2}{2g} - \frac{c^2}{2g} > p_t, \qquad (60)$$

where p_t in feet is the steam pressure corresponding to the temperature of the water in the suction pipe. If this inequality is not true, the suction column will part. Of course, the term h_1 is the one which is usually responsible for the inequality not holding, and a change in it may make this true. It may be,

though, that v_s^2 is so great and that $\frac{l}{d}$ is so great that the term $\zeta_s \frac{v_s^2}{2g}$ is responsible for this inequality not holding. In such a case the diameter of the suction must be increased, giving a smaller value of $\frac{l}{d}$ and v_s^2. The bends in the pipe must be gradual to keep the value of m as small as possible.

Although the velocity in the suction pipe should be such that there is no danger of the column of water parting, the loss should be reduced to a low value. It is difficult to tell just what velocity to allow to give the loss in the suction pipe a value consistent with the other losses. A method used by some is to make the velocity in the suction pipe equal to the velocity caused by a certain percentage of the total head. Neumann uses 2 per cent of the head kH. If the suction pipe is short and direct, this value will be used while with a long pipe (say 50 diameter) a 1 per cent value will be employed. This gives

$$v_s = 8.02\sqrt{\frac{kH}{50}} \text{ (short pipe)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (61)$$

$$= \frac{8.02}{10}\sqrt{kH} \text{ (long pipe)}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (62)$$

In any case a discussion similar to that given under the discharge pipe could be used at this point.

The loss $\zeta_{ent}\frac{c^2}{2g}$ is due to the incorrect angular relation between the fixed vanes (if used) and the movable ones at entrance and also to the interference between the moving or fixed vanes with the flow of water, causing impact. The same kind of losses occur at exit in the term $\zeta_{ex}\frac{u^2}{2g}$. For this reason these two will be considered at the same time.

In Fig. 416 a series of movable vanes is placed opposite a set of fixed vanes. If the velocities are as shown in the diagram, the water as it leaves the moving vanes is traveling

through space as shown in the figure. When this water strikes the blunt vane, there is a certain amount of loss due to sudden contraction, and moreover when the water leaves the blunt end of the moving vane there is a loss due to sudden enlargement.

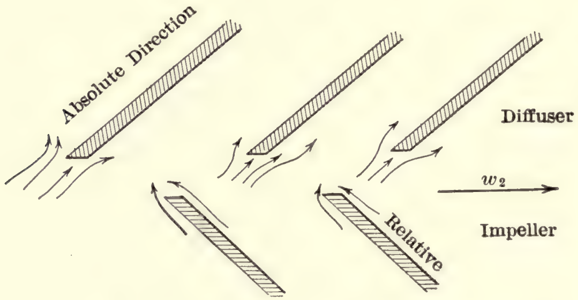


FIG. 416.—Interference of Blunt Vanes.

To cut down these losses the edges of the vanes are sharpened as shown in Fig. 417, and there should be a certain amount of clearance between the two sets of vanes. This clearance permits the water leaving one set of vanes to come together as one

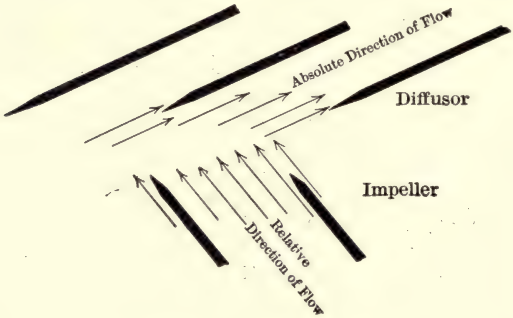


FIG. 417.—Pointed Vanes.

mass of water before entering the other set of vanes. The action of one set of vanes on the other, if the vanes are so close that there is a space between the masses of water discharging from each channel, should be such that much noise and shock would result.

By cutting the edges of the vanes and by separating the various sets, these coefficients ζ_{ent} and ζ_{ex} are made small.

Now these losses are those which occur when the relations between the various velocities are such that there is no impact. In many pumps, however, there is a change in the pressure or quantity discharged while the pump is driven at a constant speed. This means a change in the velocities c_2 and u on the outlet side, since these are given by the equations:

$$Q = A_2 c_2 = A_{ex} u.$$

If Q is increased c_2 and u will be increased, while a decrease

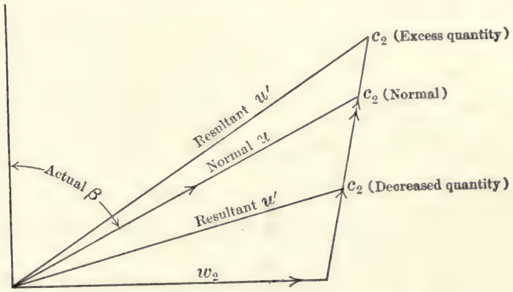


FIG. 418.—Effect of Changing Quantity with Fixed Speed.

in Q will change c_2 and u in the opposite manner. These changes are shown in Fig. 418. It is seen that the resultant of c_2 and w_2 is not in the direction of u nor of the value of u. For this reason there is a loss due to a sudden change in velocity of

$$\frac{(u-u')^2}{2g} = \text{loss},$$

and a loss due to the impact against the side of the vane which may not be very large, as has been proven by experiments on the loss due to the impact of a jet against a flat plate at right angles to it. A similar loss to this occurs at entrance when the quantity changes from the normal amount.

The losses in the vanes, $\zeta_v \frac{c_1}{2g}$, and that in the diffuser, $\zeta_d \frac{u^2}{2g}$, are similar to loss in pipes and bends. They depend on the length of the passage l, the hydraulic radius $\frac{A}{p}$ (A = area of passage, p = the perimeter of passage), the smoothness of the passage, and its radius of curvature. These losses are of the form

$$\zeta_v = kk' \frac{l}{\frac{A}{p}},$$

k, being the coefficient for straight pipes, $= \frac{.02}{4} = 0.005$, and k being the coefficient for curvature.

The loss $\frac{(u_3 - v_0)^2}{2g}$ on entrance to the volute casing around the diffuser may be eliminated by making $v_0 = u_3$. In any case it is a small quantity.

The loss $\zeta_0 \frac{v_0^2}{2g}$ is the loss which occurs in the discharge pipe and is, therefore, similar to the loss in the discharge pipe of any pump,

$$\zeta_0 = f \frac{l}{d}.$$

In the design of an installation a number of pipes should be investigated for loss and for cost. That one should be taken which gives a result such that the use of a larger pipe would increase the yearly cost of interest, depreciation, insurance and taxes on the investment, more than the saving in the decreased cost of power by the reduction in the lost head, while the use of a smaller pipe would, on account of the greater loss in head, increase the cost of power more than the saving in the yearly cost for interest, depreciation, insurance and taxes. A velocity may be assumed in terms of the total head kH as in the case of the suction pipe, using the same values; however, the method given above is the better one.

When time will not permit of this investigation, the same values may be used as for the suction, giving

$$v_0=8.02\sqrt{\frac{kH}{50}},$$

or,

$$v_0=\frac{8.02}{10}\sqrt{kH}.$$

In this work for design ζ_v' and ζ_d' may each be taken as $\frac{0.15u^2}{2g}$, and these terms will include the losses at entrance to runner and to diffuser.

Leakage

The leakage of water through the clearance space is due to the difference in pressure within the runner and on the outside. Fig. 419 shows several forms of impellers. The pressure on the inside of the runner is p_c (Fig. 406), while on the outside, the pressure is p_a. If the clearance is t feet and the circumference is $2\pi r_2$, the area through which leakage may occur is

$$2\times 2\pi r_2 t.$$

The velocity is

$$v_l=c_v 8.02\sqrt{p_c-p_a}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (63)$$

c_v = coefficient of velocity.

From Eqs. (1) to (8),

$$p_c=-\frac{w_2^2}{2g}+\zeta_{ex}\frac{u^2}{2g}-\frac{u^2}{2g}+\zeta_d\frac{u^2}{2g}+\zeta_v\frac{v_1^2}{2g}+h_2+a+\zeta_0\frac{v_0^2}{2g}+\frac{v_0^2}{2g};$$

$$p_a=a-h_1-\frac{\zeta v_s^2}{2g}-\frac{c^2}{2g};$$

$$p_c-p_a=H+(\zeta_s'+\zeta'_{ex}+\zeta_d'+\zeta_v'+\zeta_0)\frac{u^2}{2g}+\frac{v_0^2}{2g}+\frac{c^2}{2g}-\frac{u^2}{2g}. \quad . \quad (64)$$

Neumann proposes to assume $\frac{c^2}{2g}=(\zeta'_{ent}+\zeta_v')\frac{u^2}{2g}$.

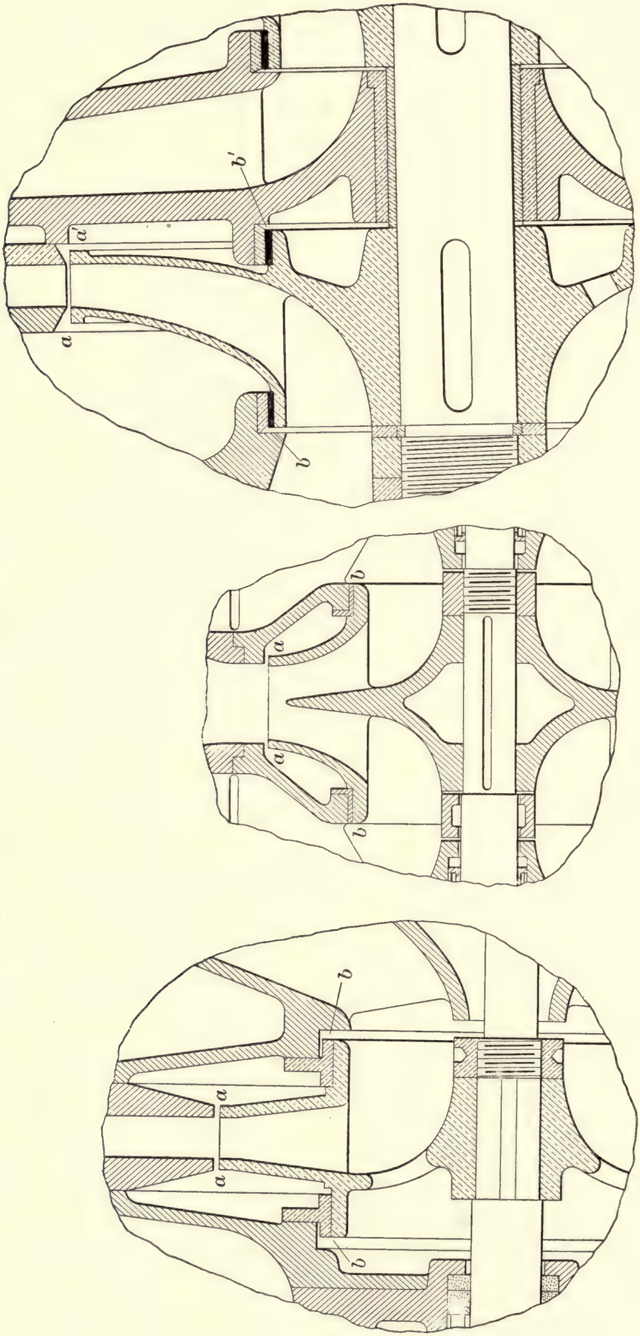


FIG. 419.—Points of Leakage.

Then $$p_c - p_a = H(1 + l + r) - \frac{u^2}{2g} = kH - \frac{u^2}{2g}. \quad . \quad . \quad . \quad (65)$$

Hence $$v_l = c_v \, 8.02\sqrt{kH - \frac{u^2}{2g}} = c_v\sqrt{2gkH - u^2};$$

$$Q_e = c_v 4\pi r_2 t \, 8.02\sqrt{kH - \frac{u^2}{2g}}. \quad . \quad . \quad . \quad . \quad . \quad (66)$$

From this it is seen that the leakage is a function of the total head, but it decreases with an increase in the absolute velocity

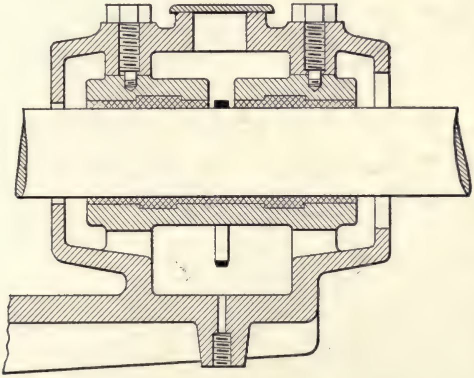


FIG. 420.—Non-aligning Ring Oiling Bearing.

of discharge from the pump. If u is small, the velocity of discharge through the clearance space is practically that due to the total head acting on the pump. For this reason it is necessary to make the clearance between the edge of the runner and the casing as small as possible. At times the leakage is diminished by placing rings on the casing and runner as at b, Fig. 419. This increases the pressure in the space a to p_a' and so decreases the flow. Such an arrangement will increase the end thrust if the runner is not a double one.

Friction at Bearings and Stuffing Boxes. The friction at a bearing may be made very small by the use of ring oiling bearings and by the use of spherical pivoted bearing boxes. Such boxes are shown in Figs. 420 and 421. With the proper

lubrication the coefficient of friction will be between 0.01 and 0.004, so that the work of friction will be

$$F=2\pi rN\mu W=0.04rNW, \quad . \quad . \quad . \quad . \quad . \quad . \quad (67)$$

W = weight of impeller or shaft for a motor-driven pump;
= resultant of weight and belt pull for a belt-driven pump;
r = radius of shaft in feet;
F = foot-pounds of work per minute due to bearing friction.

The coefficient μ for lubricated journals varies with the pressure and with the velocity. This must be taken into account

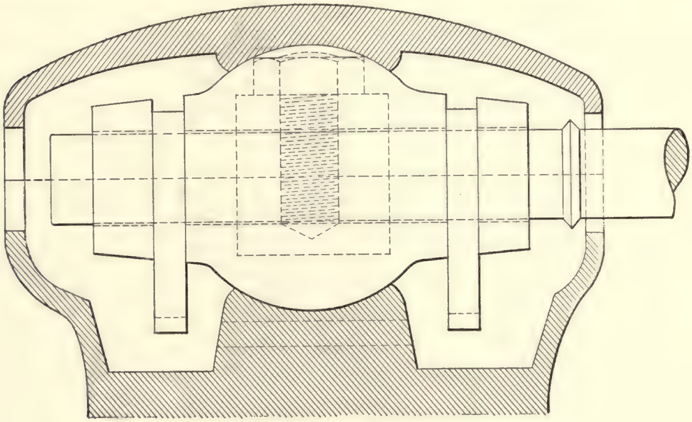


FIG. 421.—Self-aligning Ring Oiling Bearings.

in the determination of the friction. Fig. 422 shows the variation of μ with p_a, the pressure per square inch of projected area and with velocity. These curves are for a good grade of spindle oil.

In fixing the length of the bearing, the designer would assume the length from practice, as the allowance of 50 to 200 lbs. per square inch of projected area would not give a bearing long enough for a motor-driven pump. In the belt-driven pump the length is determined by the formula below, using 50 lbs. as the allowable bearing pressure,

$$l=\frac{W}{50d}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (68)$$

This friction may be increased if the bearings bind in any way, and for that reason self-aligning bearings should be used. The plain ring oiling bearing shown in Fig. 420 is often used. With the bearings in good condition the friction at these points will be a very small part of the total power, especially with motor-driven pumps. The end thrust due to the pressure action of the entering water in a single-flow pump is carried by a thrust collar bearing as shown in Fig. 423. The pressure is

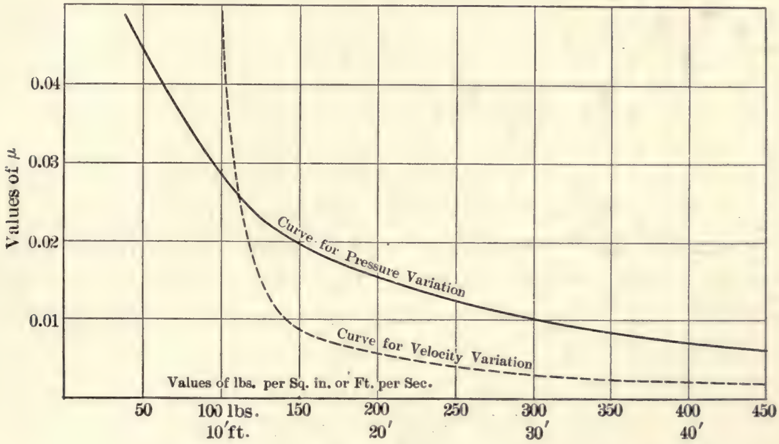


FIG. 422.—Curves of Variation of Coefficient of Friction.

represented by P_a and the necessary area is found by allowing a bearing pressure of 60 lbs. per square inch,

$$n\frac{\pi}{4}(d_1{}^2-d^2)=\frac{P_a}{60}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

The friction from this source is found by using a coefficient of friction μ which has the value 0.035 for properly lubricated thrust bearings:

$$F_{\prime\prime}=\mu P_a\pi\left(\frac{d_1+d}{2\times 12}\right)N \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (70)$$

$F_{\prime\prime}$ = work of friction per minute in foot-pounds;
d_1 = outer diameter of collars in inches;
d = diameter of shaft in inches.

The value of μ for thrust bearings does not vary much with the speed or pressure. The value of μ given above is a mean value determined by Tower in his noted experiments.

The friction from stuffing boxes depends on the amount, kind and arrangement of the packing. From experiments performed by Professor Benjamin, the formula below may be used as a guide for the friction to be expected from each stuffing box when the nuts are tightened by the application of 16 pounds at the end of a 7-inch wrench. (The bolts are spaced 6 inches apart on the pitch circle.)

$$F' = 100\, l \times v;$$

l = length of packing in feet;

v = velocity of rubbing in feet per minute.

To cut down the amount of friction from the stuffing boxes, water is sometimes allowed to enter the space around the shaft on the suction side as shown in Fig. 430, so that air is not allowed to enter the suction space. This water may be drawn in without interfering with the action of the pump and at the same time the stuffing box at the end of the shaft does not have to be tight enough to be air tight. The stuffing box on the other side of the pump is under high pressure and water tends to flow outward. This should be allowed to happen, as the small amount of leakage tends to cool the stuffing box. In operating pumps the stuffing boxes on each side should be allowed to drip, as this not only cools the shaft but indicates that the packing is not too tight.

Friction of Water on Back of Impeller. The friction of water on the back of the impeller is an important item and may amount to a large part of the loss in the pump.

Let r be the radius to a particular part of the impeller and N the revolutions per minute. If p = the friction on unit area at this point, the resistance of an elementary ring is

$$W_{,,,} = p(2\pi r dr)(N 2\pi r).$$

Now, p is proportional to the square of the relative velocity

between the water and the surface of the impeller, and since this would vary from the center to the outside,

$$p = k(2\pi N r)^2,$$

hence,

$$\left.\begin{aligned}\text{Total } W_{,,,} &= (2\pi)^4 k N^3 \int_{r_c}^{r_0} r^4 dr \\ &= k' N^3 (r_0^5 - r_c^5)\end{aligned}\right\} \quad . \quad . \quad . \quad (71)$$

Professor F. G. Hesse performed a series of experiments at the University of California on a step bearing in which the friction is about the same as that occurring on the two sides of an impeller, and derives the following formula:

$$W = 176 \times 10^{-7} N^3 D^5 \text{ ft. lb. per minute}, \quad . \quad . \quad . \quad (72)$$

from this $k' = 58 \times 10^{-5}$.

D = diameter in feet;

Stodola gives the loss from the rotation of steam turbine discs in steam as

$$N' = 0.0721\ D_1^{2\cdot 5} \left(\frac{u_1}{100}\right)^3 \gamma,$$

where $N' = HP$ due to friction;
D_1 = diameter in feet;
u_1 = peripheral velocity of disc in feet per second;
γ = weight of 1 cu.ft. of medium.

He gives this as an empirical formula from the results of Odell and from his own experiments as well as the experimental work of Lewecki.

If this is changed to a form similar to that given above with $\gamma = 62.5$ it reduces to work per minute,

$$W_{,,,} = 213 \times 10^{-7} N^3 D^{5\frac{1}{2}}. \quad . \quad . \quad . \quad . \quad (73)$$

Of course this is not intended to be used with water as the medium in which the disc is revolving, but the form is quite similar and the formula is given here for reference.

Unwin in his hydraulics describes experiments for the determination of this friction and reduces the following equation:

$$F = f\frac{2^5\pi^4 N'^3 R^5}{5}$$

$$= .003\frac{2^5\pi^4}{5}\left(\frac{N}{60}\right)^2 \frac{D^5}{2^5} N = 163 \times 10^{-7} N^3 D^5, \quad . \quad . \quad (74)$$

or $\qquad k' = 52 \times 10^{-5}.$

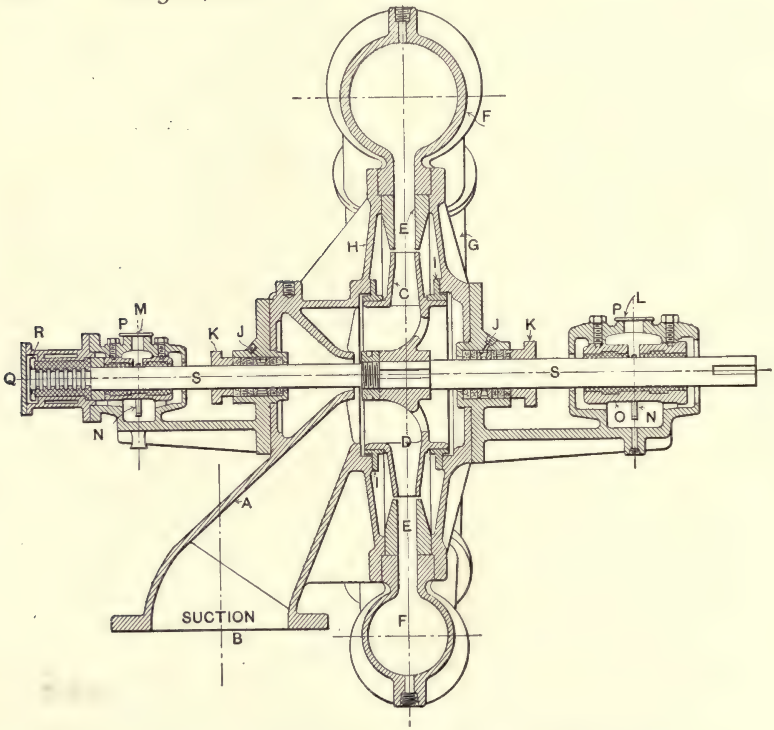


FIG. 423.—Sectional View of Worthington Volute Pump.

In order to cut down this friction loss the leakage water may be drawn off so that it will not completely fill the space between the impeller and the casing. This, however, means increased leakage, as the pressure between the wheel and the space around it at the discharge edge will be greater in this case. At times air under pressure is put in this space to keep the water back and thus reduce friction. It is well to remem-

ber that since the friction varies as N^3D^5 it is more important to use a small diameter and large N than a large D and small N when the value $\pi D_2N = w_2$ is fixed. This fact appears in practice where small impellers are found.

Forms of Centrifugal Pumps

There are many forms of centrifugal pumps. When the guide vanes are omitted the water passes directly into the volute casing through a filling ring. This was the original form of centrifugal pump. By some manufacturers it is called a **volute centrifugal pump.** Fig. 423 shows the section through a single-suction Worthington Standard volute pump. Fig. 424 shows a double-flow volute pump of one of the earlier forms. In Fig. 423, water enters the *suction head A* at the point *B* where the suction pipe is attached. From the head it enters the *impeller C*, meeting the *vanes D*, which force the water outward into the *filling ring E* and finally into the *volute casing F*. The impeller is of the form shown in Fig. 408. The vanes are enclosed on each side by a disc so that the water does not come in contact with the *bearing head G* on one side or the *suction head H* on the other. The form of impeller in Fig. 424 is not of the enclosed type and consequently there is considerable water friction. Moreover, the mechanical friction is greater in the unenclosed type, and as the vanes fit so closely to the heads a slight amount of end play will cause rubbing.

In the impeller there is an increase of energy due to the increase in the peripheral speed. This increase of energy may take the form of increase in pressure if u is the same as c, or it may be in the form of increase in kinetic energy if $p_a = p_d$. In general there is an increase in both the kinetic energy and the potential energy or pressure. In general $p_a < p_d$, so that there is leakage from the outlet of the impeller into the space around this part of the pump.

The *bushing rings I, I*, prevent excessive leakage from the space around the impeller into the suction space. The space within the right-hand bushing ring is connected to the suction

space by the openings in the central part of the impeller disc, so that any leakage at the ring will be taken into the suction space. These openings also equalize the pressure at the center of the wheel and reduce the end thrust. The stuffing boxes at the two sides of the pump are equipped with a *cage J* between the two sets of packing rings for the introduction of oil or grease from cups attached to the stuffing boxes, *K*. The *outboard bearing L* and the *suction bearing M* are provided with *oil rings*

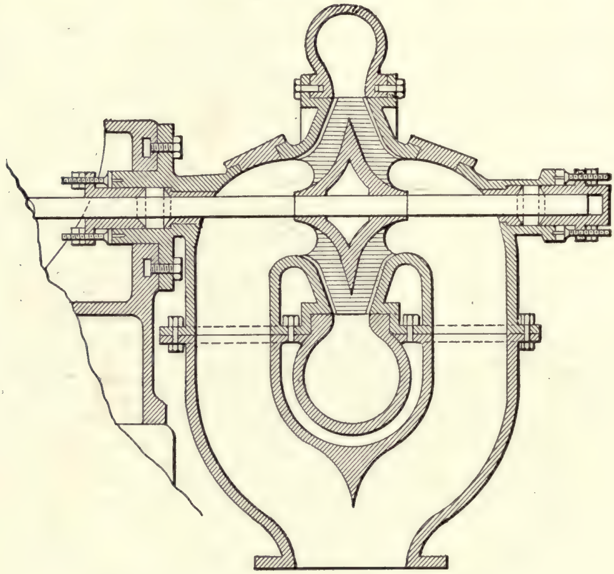


FIG. 424.—Double Flow Unincased Impeller.

N and the *oil cellar O*. The oil is introduced through the opening capped by the cover *P*. The *thrust bearing Q*, supporting the small end thrust, is principally used to keep the impeller in a central position. The cap *R* on the end of the shaft is an *oil thrower*. The *shaft S* is supported by two bearings and turned with a small shoulder at the center so that the impeller may be secured in a given longitudinal position by the *nut*. The impeller is keyed to the shaft and the coupling used to attach this to the motor is also keyed. The ribs around the volute casing are used to strengthen that casting, which is not a com-

plete pipe on account of the opening from the filling ring, and for that reason the bending action on the casting requires additional ribs.

It is the object of the whirlpool chamber to reduce the absolute velocity of discharge, changing it into pressure. It does not accomplish this efficiently, as there is shock here and a sudden change in velocity which means a loss. By having this chamber gradually increase in area the velocity of the water decreases as it passes through, and with this action the

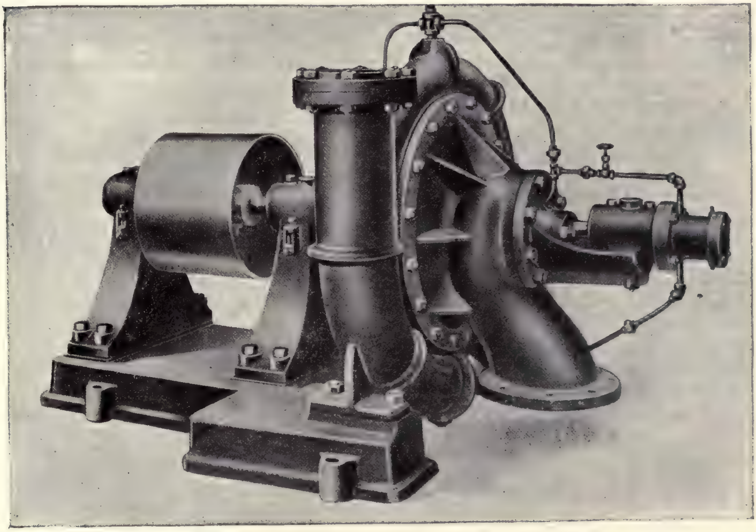


FIG. 425.—Worthington Belted Volute Pump.

pressure increases. To make this change in an effective manner, however, the diffuser is used, in which there is no shock, and the velocity head is changed into pressure.

Fig. 425 illustrates one of the Worthington single-flow volute pumps for a belt drive, while Fig. 426 illustrates a double-flow volute pump, motor driven. These figures show the external appearance of the pumps; the pipes for water circulation in the stuffing boxes and thrust bearing; the stiffening webs; the method of supporting the casing and the method of attaching the heads to the casing. The double-flow pump

of Fig. 426 is not equipped with a thrust bearing, as in this type of pump there should be no end thrust.

These pumps are intended primarily for heads up to 65 feet, and in most cases when this head is exceeded the turbine type of pump with a set of diffuser vanes would be used. The impellers built by Worthington are specially designed to suit the conditions of the particular service required by the purchaser, and this should always be done, as each speed and head

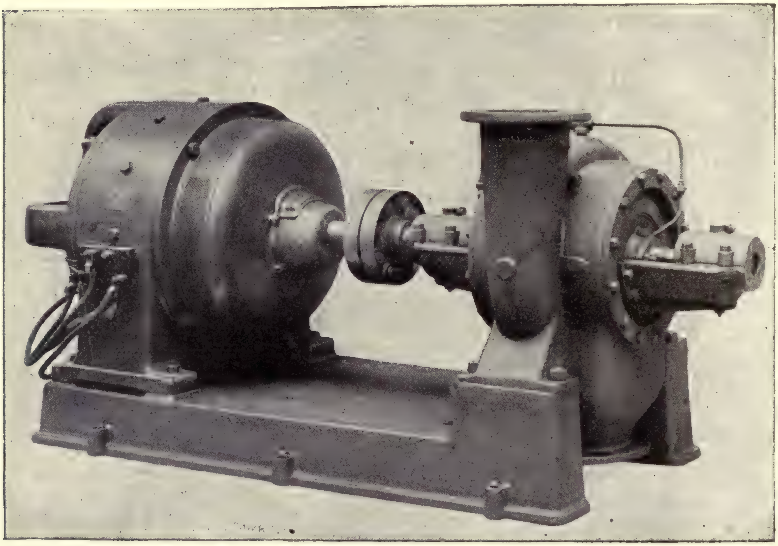


Fig. 426.—Motor Driven Worthington Double Flow Volute Pump.

requires a definite form of impeller. The suction head shown in Fig. 423 is so arranged that water may enter from all parts, giving an adequate supply to the impeller.

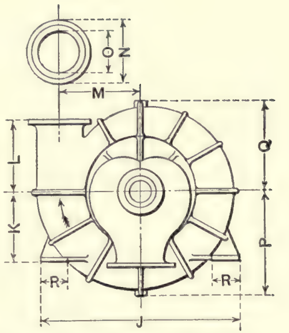
These pumps are built in different sizes, as shown by the tables below; the first table giving the capacities, speeds, heads and power; the second, the dimensions of the casing.

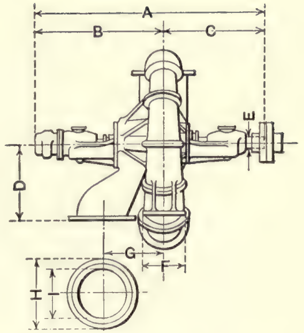
The pumps shown so far have been arranged with the suction below, while the discharge has been taken off vertically. This arrangement may be changed to suit as shown in Fig. 427, which illustrates a few of the different arrangements possible with their designating numbers.

TABLE OF CAPACITIES, SPEEDS AND POWERS FOR WORTHINGTON STANDARD VOLUTE PUMPS

Size	Capacity. Normal.	Capacity. Maximum	Head	5 ft.	10 ft.	15 ft.	20 ft.	25 ft.	30 ft.	35 ft.	40 ft.	45 ft.	50 ft.	55 ft.	60 ft.	65 ft.
$1\frac{1}{2}$	45	60	Speed	500–810	680–1080	780–1170	870–1305	950–1370	1020–1550	1120–1650	1260–1740	1400–1845	1480–1935	1580–2025	1660–2085	1720–2160
			Power	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{8}$	$2\frac{1}{2}$
2	75	100	Speed	420–720	570–925	650–1000	725–1125	790–1225	870–1325	940–1415	1050–1495	1160–1555	1230–1655	1320–1750	1380–1780	1430–1855
			Power	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	4
$2\frac{1}{2}$	125	150	Speed	360–630	485–825	560–875	625–980	680–1080	740–1160	800–1240	900–1305	1000–1385	1050–1450	1125–1520	1190–1565	1230–1630
			Power	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	3	$3\frac{1}{2}$	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{3}{4}$	5	$5\frac{1}{2}$
3	200	250	Speed	315–540	425–735	490–780	545–875	590–955	650–1035	700–1100	790–1160	875–1235	925–1285	940–1350	1040–1395	1080–1450
			Power	$1\frac{1}{4}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	$3\frac{3}{4}$	$3\frac{3}{4}$	$4\frac{1}{2}$	5	$5\frac{3}{4}$	$6\frac{1}{4}$	7	$7\frac{1}{2}$	$8\frac{1}{4}$
4	350	450	Speed	280–450	375–710	435–765	485–855	525–940	580–1015	625–1100	700–1135	775–1170	820–1240	880–1305	920–1325	960–1385
			Power	2	$2\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{1}{2}$	$5\frac{1}{4}$	$6\frac{1}{4}$	$7\frac{1}{4}$	$8\frac{1}{4}$	$9\frac{1}{4}$	$10\frac{1}{4}$	$11\frac{1}{4}$	$12\frac{1}{2}$	$13\frac{1}{2}$
5	600	700	Speed	250–385	340–595	390–675	435–735	475–810	520–855	560–910	630–955	700–1010	735–1050	790–1080	830–1115	860–1160
			Power	$2\frac{3}{4}$	4	5	$5\frac{1}{2}$	7	$8\frac{1}{4}$	$9\frac{3}{4}$	$10\frac{3}{4}$	$12\frac{1}{4}$	$13\frac{3}{4}$	$15\frac{1}{4}$	$16\frac{1}{4}$	$17\frac{3}{4}$
6	800	1000	Speed	230–315	310–510	355–575	395–630	430–685	475–735	510–780	570–820	635–855	670–890	720–925	755–955	780–990
			Power	$3\frac{3}{4}$	$5\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{4}$	9	11	$12\frac{1}{2}$	$14\frac{1}{2}$	$16\frac{1}{4}$	18	20	$21\frac{1}{2}$	$23\frac{1}{2}$
8	1500	1800	Speed	195–260	260–420	300–475	335–525	365–570	400–620	435–640	485–675	540–710	570–740	610–765	640–800	660–820
			Power	$5\frac{3}{4}$	9	$11\frac{1}{2}$	13	16	20	23	26	$29\frac{1}{2}$	$32\frac{1}{2}$	36	39	42
10	2500	2800	Speed	180–225	245–400	280–450	310–490	340–525	370–555	400–585	450–620	500–640	525–655	565–675	590–700	615–720
			Power	8	13	$16\frac{1}{2}$	20	25	30	35	40	$45\frac{1}{2}$	$50\frac{1}{2}$	$55\frac{1}{2}$	$60\frac{1}{2}$	$65\frac{1}{2}$
12	3500	4000	Speed	168–180	225–370	260–415	290–450	315–475	345–500	375–530	420–550	465–575	490–600	525–620	550–645	570–655
			Power	11	17	22	29	37	44	50	58	65	78	80	87	94
14	5000	5500	Speed	158–170	210–340	245–380	270–410	295–440	325–465	350–485	395–510	435–530	460–555	495–565	520–600	540–625
			Power	$14\frac{1}{2}$	23	30	40	50	60	70	79	89	99	109	119	129
16	6500	7500	Speed	148–160	200–325	230–365	255–395	280–425	305–445	330–465	370–495	410–515	430–530	465–550	485–570	505–585
			Power	18	32	40	54	68	81	95	108	122	135	149	162	175
18	8000	9500	Speed	134–145	180–295	205–330	230–355	250–380	275–400	295–425	330–440	365–460	390–475	415–495	435–510	450–525
			Power	24	40	51	69	86	102	119	137	155	171	188	205	218
20	10000	12000	Speed	120–135	162–250	185–315	205–325	225–340	245–365	270–380	300–395	335–415	350–435	375–445	395–460	410–475
			Power	30	50	65	86	108	130	150	172	193	215	236	258	280
24	15000	18000	Speed	105–125	142–235	162–265	180–290	200–310	215–325	235–340	260–355	290–370	305–380	330–400	345–410	360–425
			Power	46	76	97	130	162	194	227	259	293	324	357	390	420

NOTE.—Pumps may be designed for any speeds within the range given in the above table, but when a certain speed has been selected and the pump designed accordingly, no change in speed can be made without affecting the capacity and head. The powers given are figured for the maximum quantity of water. The power required for handling less water than the maximum is approximately in proportion to the quantity when the quantity is not less than in the maximum capacity for the next smaller size pump. Motors should be selected of somewhat larger capacity than the horse-power given in the table, to allow for possible overload.





SIZE	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R
2	$27\frac{11}{16}$	$12\frac{9}{16}$	$15\frac{1}{8}$	$8\frac{1}{2}$	1	4	$5\frac{1}{2}$	7	$2\frac{1}{2}$	$19\frac{3}{4}$	7	$8\frac{1}{2}$	$8\frac{1}{2}$	6	2	$9\frac{3}{4}$	9	$2\frac{1}{2}$
$2\frac{1}{2}$	$30\frac{1}{4}$	$13\frac{1}{4}$	17	$10\frac{1}{4}$	$1\frac{1}{4}$	5	$7\frac{1}{16}$	$7\frac{1}{2}$	3	$23\frac{1}{2}$	$8\frac{1}{2}$	$9\frac{1}{2}$	10	7	$2\frac{1}{2}$	$11\frac{1}{2}$	$10\frac{9}{16}$	$3\frac{1}{2}$
3	$31\frac{7}{8}$	$14\frac{9}{16}$	$17\frac{5}{16}$	12	$1\frac{1}{4}$	6	$7\frac{5}{8}$	9	4	$26\frac{1}{2}$	10	$10\frac{1}{2}$	$11\frac{5}{8}$	$7\frac{1}{2}$	3	$13\frac{5}{16}$	$12\frac{3}{16}$	4
4	$40\frac{9}{16}$	$19\frac{5}{8}$	$20\frac{15}{16}$	$12\frac{1}{2}$	$1\frac{1}{2}$	7	8	10	5	$32\frac{1}{2}$	$10\frac{1}{2}$	12	$13\frac{1}{4}$	9	4	$16\frac{9}{16}$	$15\frac{1}{8}$	$4\frac{1}{4}$
5	$44\frac{1}{16}$	$23\frac{1}{16}$	21	16	$1\frac{3}{4}$	7	$10\frac{1}{4}$	11	6	$32\frac{1}{2}$	$13\frac{1}{2}$	13	15	10	5	$18\frac{13}{16}$	17	$4\frac{1}{4}$
6	$52\frac{15}{16}$	$28\frac{9}{16}$	$24\frac{3}{8}$	16	2	9	12	$13\frac{1}{2}$	8	37	15	$14\frac{1}{2}$	$16\frac{3}{4}$	11	6	$20\frac{13}{16}$	$18\frac{5}{8}$	5
8	$55\frac{1}{8}$	$28\frac{5}{16}$	$26\frac{13}{16}$	$18\frac{1}{2}$	$2\frac{1}{4}$	$9\frac{1}{2}$	$13\frac{3}{4}$	16	10	48	16	18	$20\frac{1}{4}$	$13\frac{1}{2}$	8	$25\frac{3}{8}$	$22\frac{1}{2}$	5
10	$67\frac{3}{16}$	$36\frac{15}{16}$	$30\frac{1}{4}$	22	$2\frac{3}{4}$	$10\frac{1}{2}$	$16\frac{3}{4}$	19	12	55	$18\frac{1}{2}$	19	$22\frac{1}{2}$	16	10	$28\frac{3}{4}$	$25\frac{1}{8}$	$6\frac{1}{2}$
12	$75\frac{5}{16}$	$40\frac{5}{16}$	35	24	$3\frac{1}{4}$	12	$19\frac{1}{4}$	21	14	64	21	24	$24\frac{3}{4}$	19	12	$31\frac{7}{8}$	$27\frac{1}{2}$	8
14	$77\frac{13}{16}$	$43\frac{5}{8}$	$34\frac{3}{16}$	26	$3\frac{1}{2}$	14	$20\frac{7}{8}$	$23\frac{1}{2}$	16	67	24	24	27	21	14	$35\frac{1}{4}$	$30\frac{1}{8}$	9
16	$80\frac{3}{8}$	$41\frac{7}{16}$	$35\frac{15}{16}$	32	$3\frac{1}{2}$	16	24	25	18	76	27	27	$30\frac{1}{2}$	$23\frac{1}{2}$	16	$39\frac{7}{8}$	34	10
18	$78\frac{15}{16}$	$39\frac{13}{16}$	$39\frac{1}{8}$	23	$3\frac{1}{2}$	18	22	$27\frac{1}{2}$	20	$76\frac{1}{2}$	29	33	32	25	18	$40\frac{1}{2}$	$35\frac{1}{2}$	10
20	$90\frac{3}{4}$	48	$42\frac{3}{4}$	38	$4\frac{1}{2}$	20	$29\frac{1}{4}$	$29\frac{1}{2}$	22	92	32	34	36	$27\frac{1}{2}$	20	$47\frac{3}{16}$	$39\frac{13}{16}$	12
24	$93\frac{7}{16}$	$47\frac{13}{16}$	$45\frac{5}{8}$	27	5	24	$29\frac{1}{2}$	$34\frac{1}{4}$	26	108	29	36	42	32	24	$54\frac{5}{8}$	$45\frac{7}{8}$	12

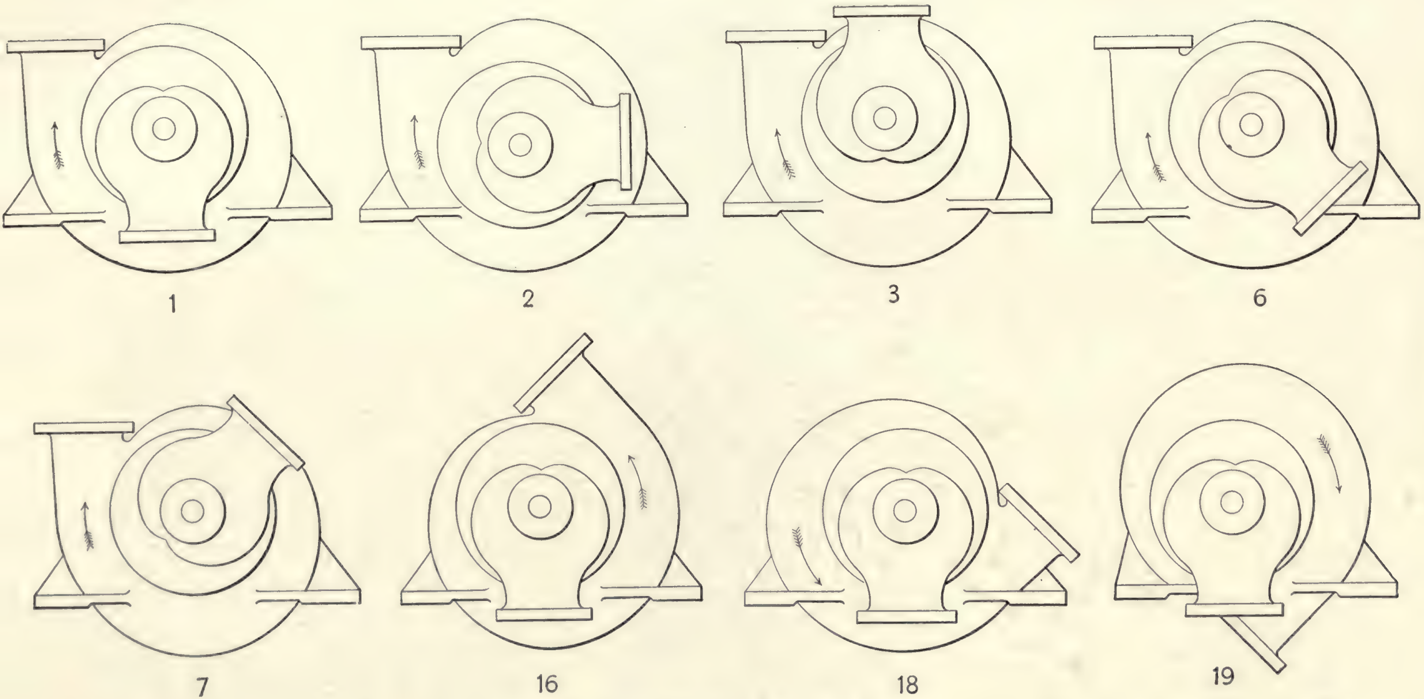


FIG. 427.—Arrangement of Pumps for Pipes.

A recent design of volute pump for a low head but large capacity is shown in Fig. 428. This pump is known as a **tri-rotor volute pump.** It was installed in the Interborough Rapid Transit Station for circulating the water through the surface condensers. It is in reality three double-flow volute pumps with four vertical suction pipes rising to the center of the casing and threading among the three horizontal discharge pipes which unite and discharge from one large opening at the front of the bed plate. The head being small, there are no stiffening ribs found on the exterior of the casing. The ring oil bearings, thrust bearing, water-supply pipes for stuffing boxes

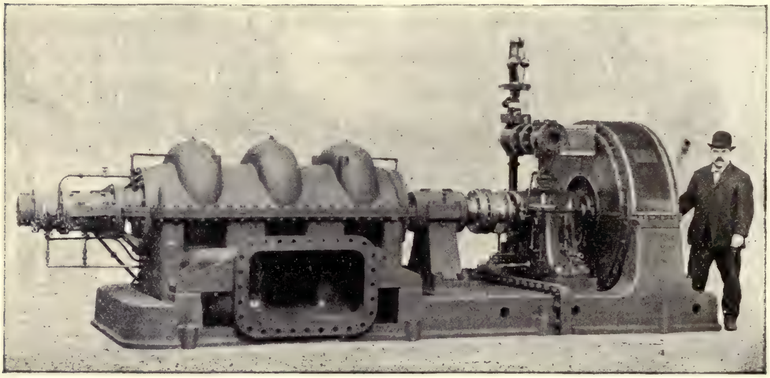


FIG. 428.—Worthington Turbine Driven Tri-rotor Volute Pump.

and thrust bearing as well as the general arrangements for casting the casing and heads are clearly seen. The figure also illustrates another method of driving the pump by a steam turbine.

The section, Fig. 429, is given not only to show the method of bringing the water to the various suction chambers and taking it from the separate volute casings into the one discharge opening, but the arrangement of the impeller is to be noted. The impellers here are the typical double-flow volute runners and the bushing rings, filling sleeves and bearings are all in evidence. These two figures illustrate the type of **split-casing pumps**, which type is usually employed in sugar factories, or gas plants where the pump may require cleaning frequently.

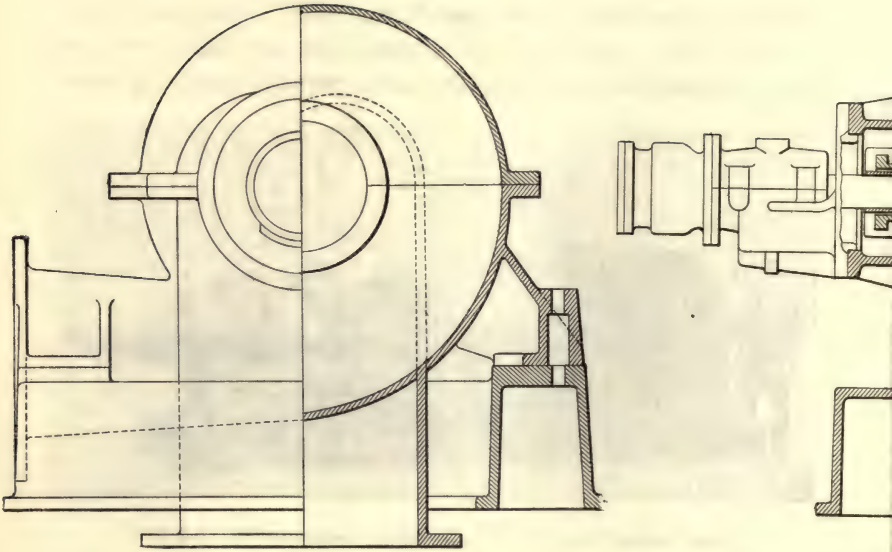
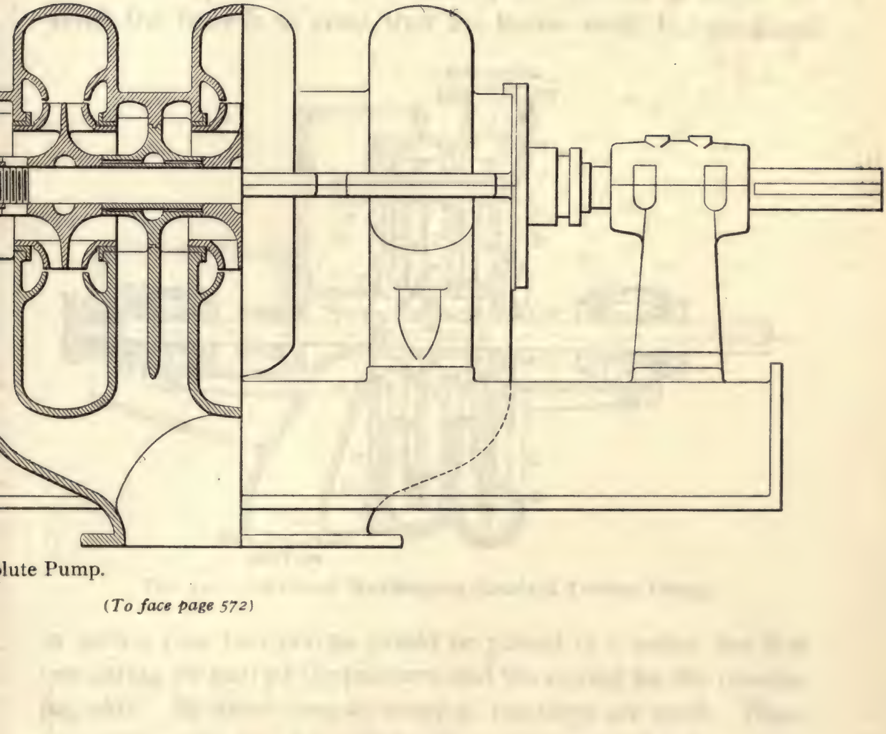


FIG. 429.—Worthington Tri



lute Pump.

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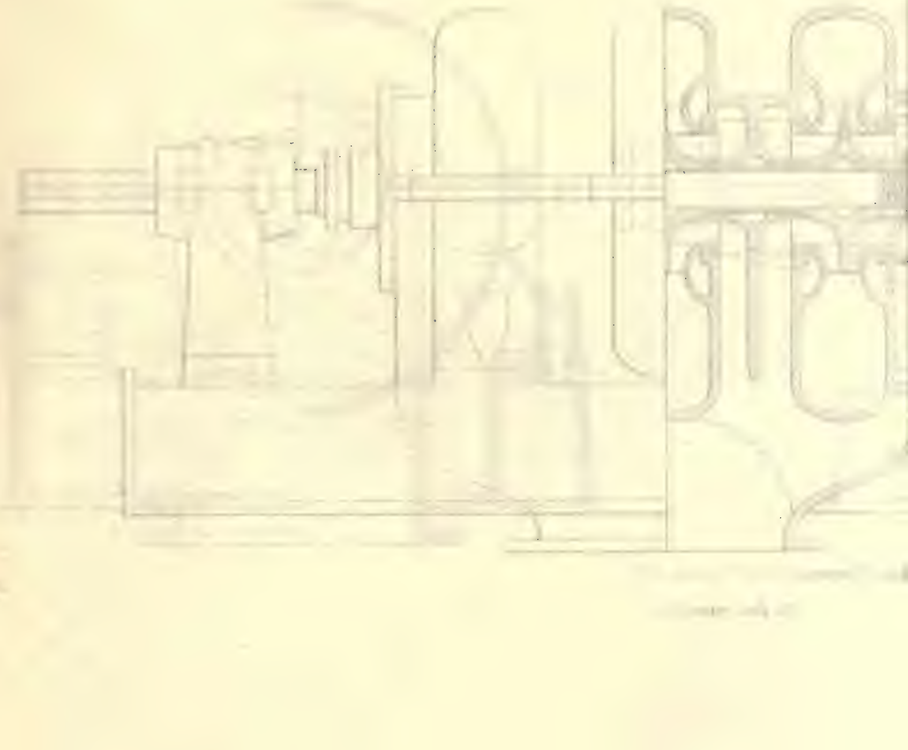


When higher heads than 60 or 70 feet are to be overcome the volute pump is replaced by the form of pump used in the development of the theory of centrifugal pumps, one with diffuser vanes. This type of pump is known as a turbine pump. The rotative speed of these pumps is often quite high, as the diameter of the impeller may then be made small. The discharge velocity in these pumps is quite high and hence to change this over to pressure head efficiently the diffuser is used. At times the head is so great that the losses would be large, and

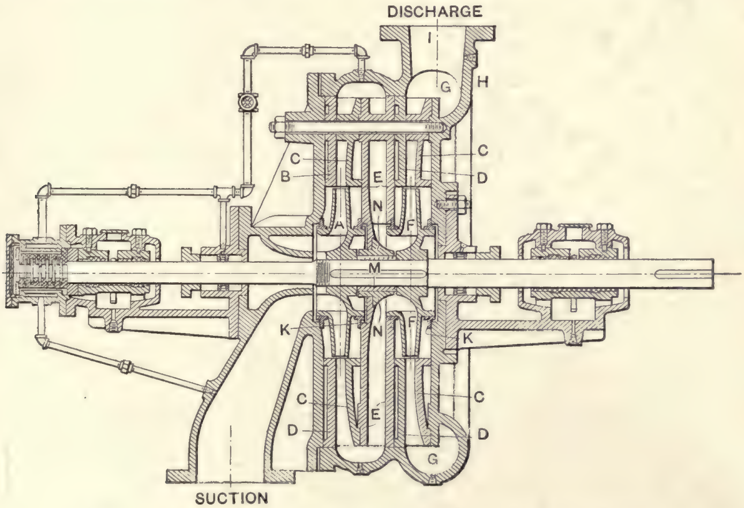


FIG. 430.—Section of Worthington Standard Turbine Pump.

in such a case two pumps would be placed in a series, the first one caring for part of the pressure and the second for the remaining part. In some cases as many as ten steps are used. These separate pumps may be combined in one casing, giving a multi-stage turbine pump, each stage caring for a head of from 75 to 200 feet. Fig. 430 is a section of a Worthington two-stage pump. The action of this pump is similar to the one described earlier, except that as the water leaves the impeller A it enters the diffuser B, which has channels formed as shown in Fig. 431. The diffuser is made up of two parts, the *diffusion ring* C with vanes and the diffusion ring without vanes D. In this

the velocity is reduced so that water passes through the *channel ring E* with a low velocity but with considerable pressure, entering the second impeller F, which discharges into a second

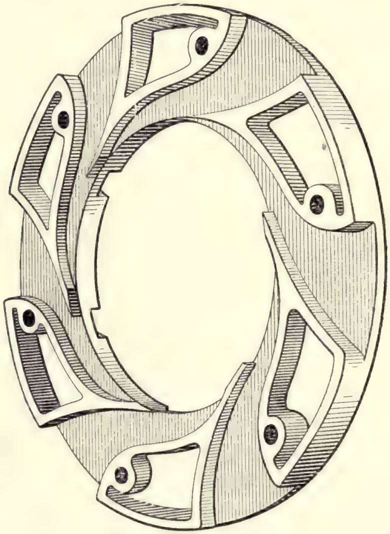


FIG. 431.—Diffuser.

diffusion ring from which it discharges into the final discharge channel G in the casing H of the pump, leaving the discharge I. The casing is cast with the outboard head solid and it is machined on the inside so that the diffusion rings and

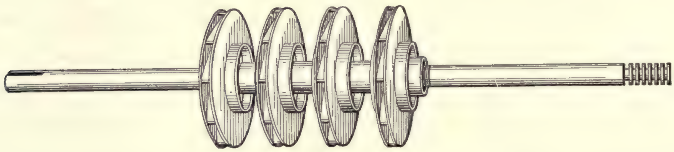


FIG. 432.—Four Impellers and Shaft.

channel rings can be introduced from one side with the suction head bolting into position on account of the turned projection on its inner face. All parts are thus held in their proper positions. Such a construction makes it possible to dismantle the pump readily for examination and repair and yet reassemble

with all parts coming into their proper positions. From Fig. 431 the manner of forming the diffusion ring with vanes is seen and it is evident that it is possible to polish these smooth to reduce the friction losses. Fig. 432 shows the four impellers of a four-stage pump. The entrance channels are seen on the right-hand sides of the discs. At the end of the shaft are seen the collars of the thrust bearing. In Fig. 430 the stuffing boxes are shown attached to the heads by flanges and the bearings are seen on heavy brackets. The thrust bearing is made in halves and bolted to the end of the suction bearing. This construction, shown in Fig. 433, is necessary for the introduction of the shaft. The cooling of this bearing is accomplished by a current of water through the shell, while continuous lubrication is effected by the oil thrower *J*. The manner of introducing water into the suction stuffing box is evident from the figure. The bushing rings at *K* prevent the excessive leakage of water into the suction.

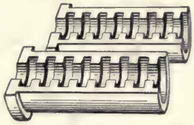


FIG. 433.—Worthington Thrust Bearing.

The method of filling the space between two impellers by means of the distance bushing *M* and the collar *N* of the channel ring may be seen in Fig. 430. The construction of a stage pump of greater number of stages would be similar to this.

The materials used for the various parts of the pump depend on the kind of liquid handled. The impellers are of cast iron or bronze and are finished smooth on the outside and inside. They are balanced. The diffusion rings are made of the same materials as the impeller. The shaft is made of steel or Tobin bronze.

Fig. 434 illustrates the method of constructing a two-stage pump as made by the Alberger Pump Company. The arrangement of the diffusing vanes, the channel ring, the bearings, the stuffing boxes and the method of uniting the various parts together are all evident from the figure. The pumps built by this company are also designed for the actual conditions of operation and the surfaces of the impeller and diffuser vanes are highly finished all over to reduce the losses.

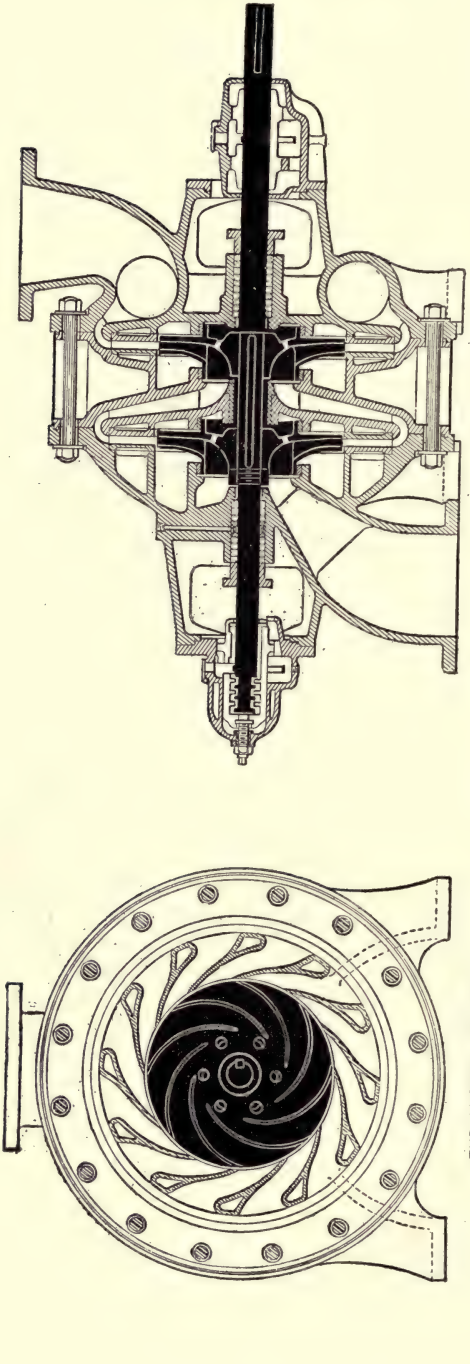


FIG. 434.—Section of Alberger Multi-stage Turbine Pump.

Fig. 435 illustrates a ten-stage motor-driven pump while Fig. 436 is an eight-stage pump. These pumps are for high heads.

When high heads are to be overcome large diameters of

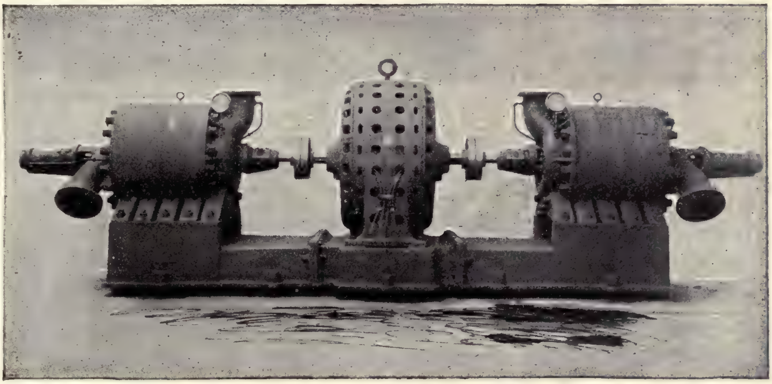


FIG. 435.—Worthington Motor Driven Ten-stage Pump.

impellers could be used to get the necessary velocity, but the friction of the water in the impeller passages and on the impeller back would be very great and the width of passages

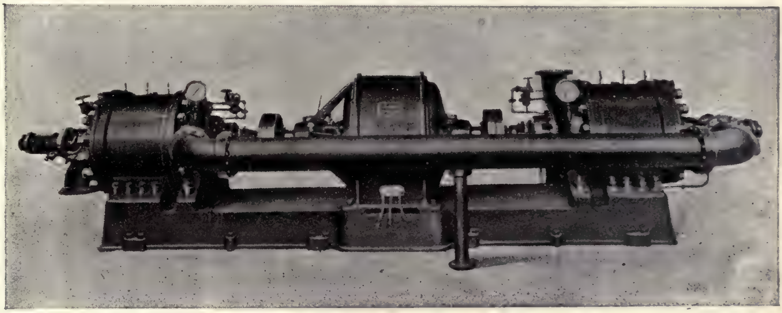


FIG. 436.—Worthington 8″ Eight-stage Turbine Mine Pump with 450 H.P. Motor.

at discharge would be very small, and moreover there would be erosion of the vanes by the water under high velocity. For these reasons about 150 feet is the limit for each stage and for great heads a large number of impellers is used. Since these take

up considerable room on the shaft there is a limit to the number which can be put between bearings without making the shaft very large. The shaft must be designed with regard to the critical speed. Five or six stages are found within one casing, but when more are required two casings are used connected in series.

It is well to think of the first wheel increasing the kinetic energy and the potential energy, then the first diffuser changes

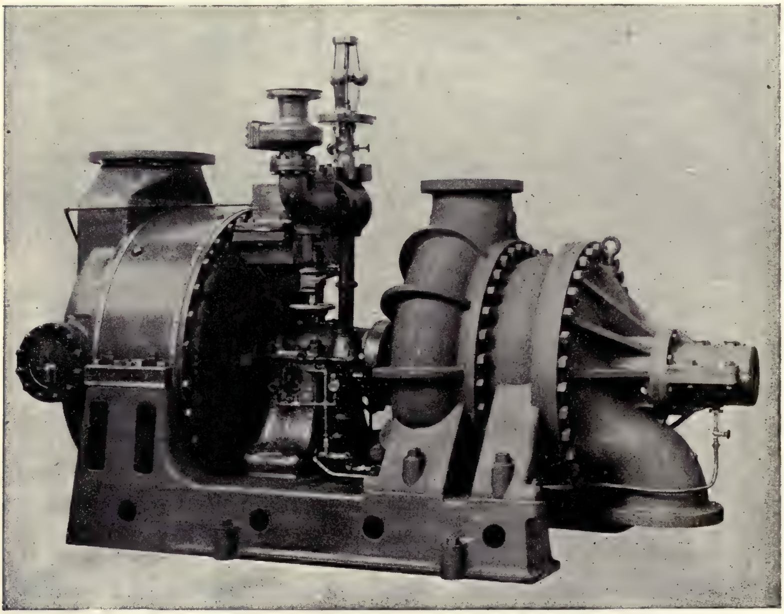


FIG. 437.—Worthington Fire Boat Turbine Pump.

the increase of kinetic energy into pressure, so that when the second impeller is entered the water has the same kinetic energy as it has at entrance to the first impeller, but with a great increase of pressure. This is repeated in each stage and at the discharge from the pump at the last stage there may be less kinetic energy than at entrance, but the potential energy in the form of pressure is very great.

When heads of several hundred feet are required for fire service the pump takes the form shown in Fig. 437. This type

of unit is installed on the New York and Baltimore fire boats.

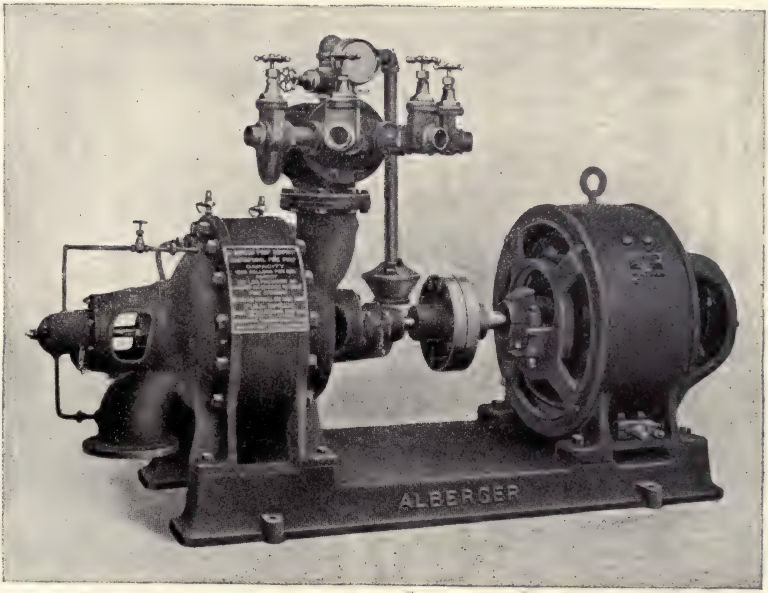


FIG. 438.—Alberger 1000-gal. Underwriter Fire Pump, Turbine Type.

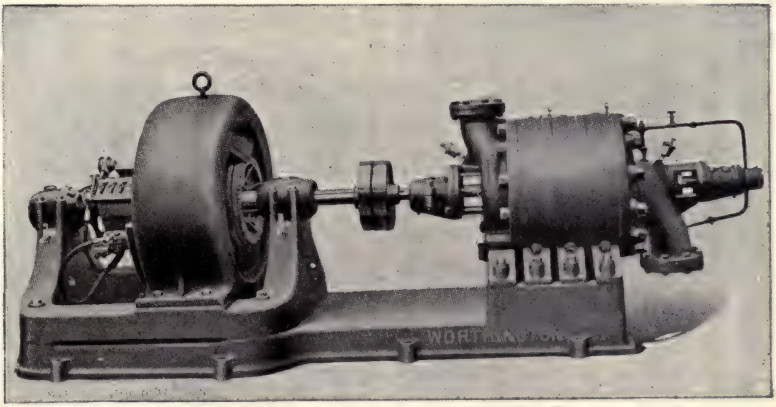


FIG. 439.—Worthington Four-stage Boiler Feed Pump.

These pumps are also used as fire pumps for factories and the Board of Fire Underwriters has issued specifications of the

same nature as those given in Chapter XI for reciprocating pumps covering the equipment and construction of centrifugal underwriter fire pumps. Fig. 438 illustrates a 1000-gallon pump of this type. The fixtures of this pump are quite similar to those discussed earlier in the work.

For boiler feeding a motor driven or steam turbine driven stage pump may be used. When turbine pumps are used there is an absence of knocking and shock common with reciprocating pumps because of the steady discharge, and moreover, as will be seen later, there is no danger of excessive pressure, even if all feed valves are closed on the boilers. Fig. 439 illustrates a four-stage boiler feed pump. Pumps similar to this could be used for house or elevator service, being controlled by a switch operated by a float or pressure regulator.

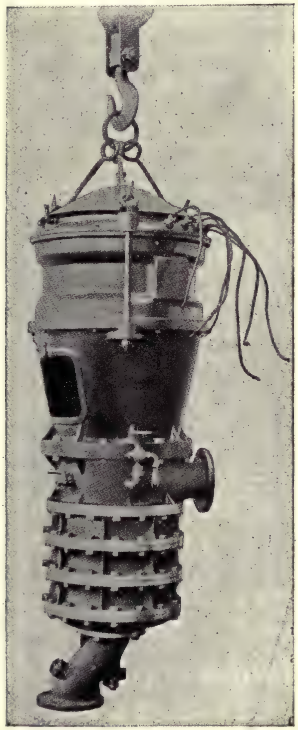
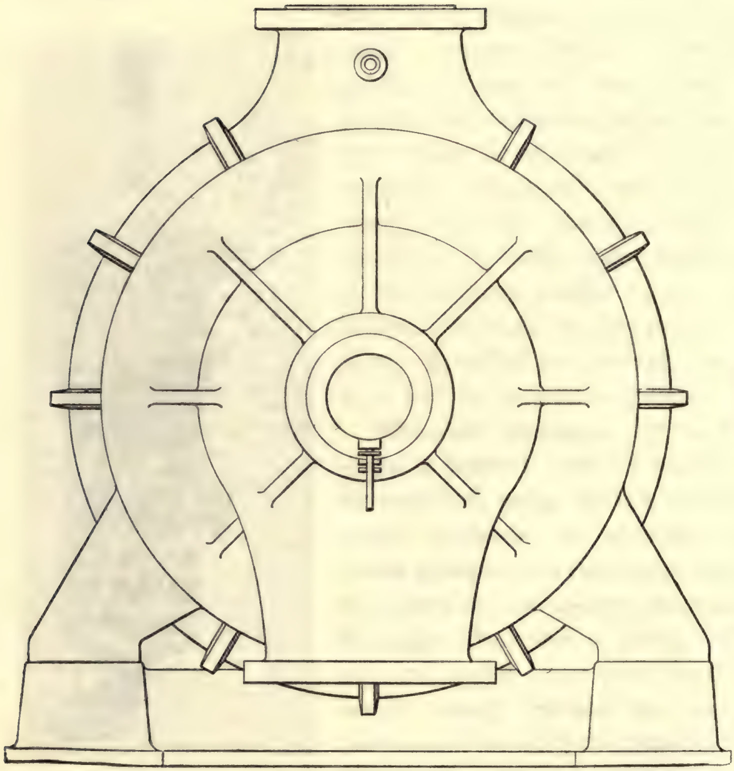


FIG. 440.—Worthington Turbine Sinking Pump.

For mine sinking or for use where space is limited, vertical pumps may be installed, using electric motors or steam turbines to operate them. These pumps have enormous capacity for their size, and operate successfully. Fig. 440 illustrates a pump built to operate under 1250 feet head in a single casing. Should the water to be handled be acid the pump is built of a composition to resist this.

When high-speed pumps are built these are made with double runners to eliminate end thrust. Fig. 441 shows a section through one of these as built by Worthington and shown in Fig. 437. In this pump the suction water for one side is carried through the open spaces between the channels of the diffuser ring of Fig. 431 and enters a well-rounded cavity leading to



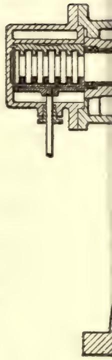
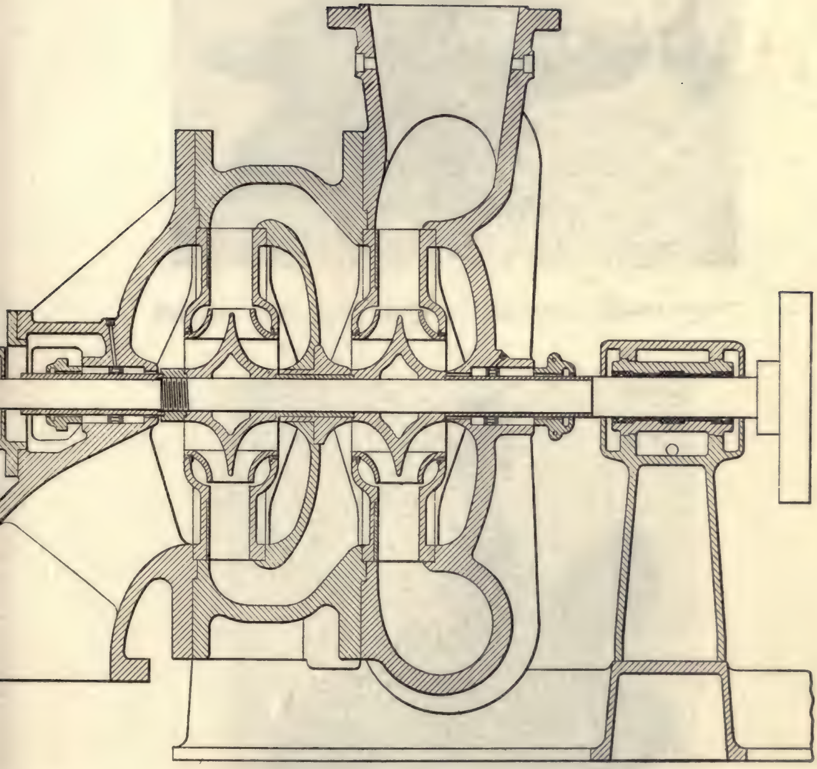


FIG. 441.—Worthington Hi



ed Pump.

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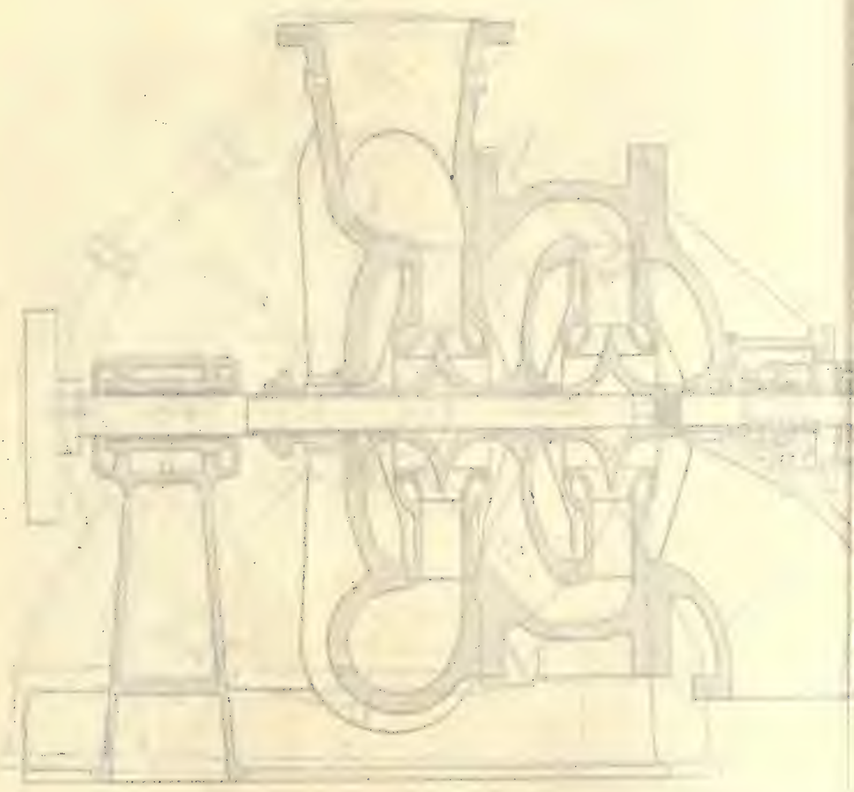

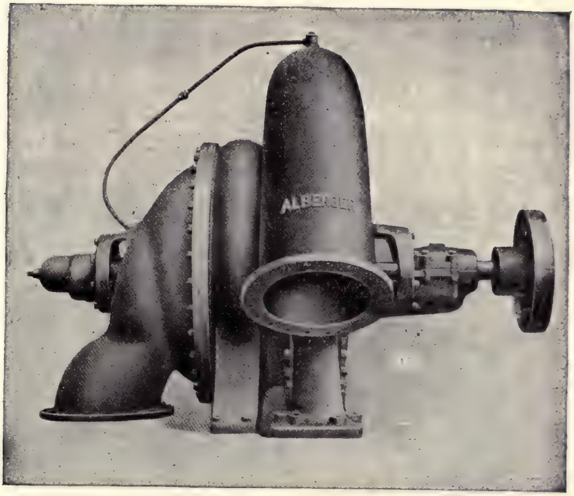

FIG. 442.—Alberger Two-stage Volute Pump, Engine Driven.

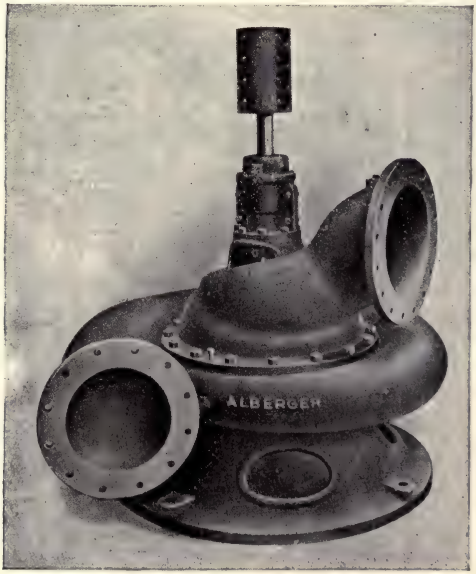

FIG. 443.—Alberger Standard Volute Pump, Vertical Shaft.

the impeller. In this way the impeller receives water on each side. The water from the diffuser is then delivered to the suction of the second stage and is delivered from the diffusion ring of this stage into the discharge channel. The velocity head in this channel is so small compared with the pressure head that there is not the need of forming the usual volute to reduce the losses and so the discharge is taken up on each side of the diffuser to a discharge flange in the center in a concentric channel.

Fig. 442 illustrates a two-stage volute pump built by Alberger. In the figure the discharge is taken from the casing at an angle to the horizontal. The general lines of the pump, the feet for attachment to bed plate, the water seal for the suction stuffing box and the well-supported bearings are to be noted. The volute pump is not used so often in stages, but there is no reason why this cannot be done, as is shown in the illustration. The volute pump may be applied with the shaft in a vertical position as shown in Fig. 443. Such a construction may be necessary on account of the lack of room, or the motor may be placed at considerable distance above the pump on account of flooding.

Another use for the centrifugal pump in the last few years is in connection with the water removal from jet condensers. Fig. 444 shows a steam-engine driven centrifugal pump attached at the lower end of an Alberger condensing head. The condensed steam and condensing water are removed and discharged into the atmosphere by the pump. The air is removed from the head by means of a dry air pump, similar to that described in Fig. 368.

In order to eliminate the end thrust with turbine pumps of several stages Mr. C. W. Larner has recently patented the method shown in Fig. 445. In this arrangement the water from the second diffusion ring is passed between the channels of the first channel ring on its passage to the second channel ring. In the figure there are four stages. The method used here is somewhat similar to that shown in the high-speed pump, Fig. 441, although in that figure each impeller is balanced by being a double-flow impeller and half of the water for the second stage

is carried through the channels in the diffusers. In Fig. 445, the impellers are single-flow impellers; consequently with the same openings at the suction side the pump will handle less water through a greater head than that for the pump shown in Fig. 441.

The method used by Sulzer to accomplish this is shown

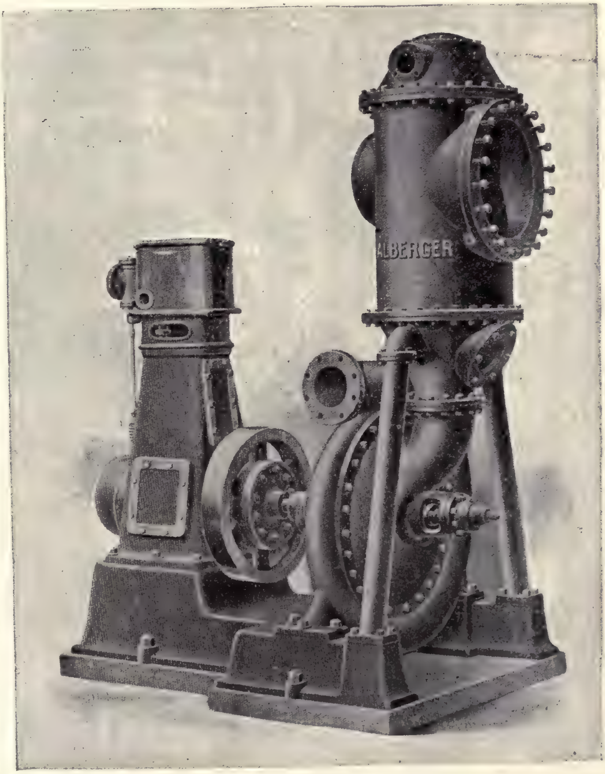

FIG. 444.—Alberger Centrifugal Condenser.

in Fig. 446. This has been used for many years. The method is to take the discharge from the first stage and pass it through the open spaces in the two diffusers, Fig. 431. The water is then brought in on the opposite side of the second impeller from that used on the first. This balances the pressure due to impact, since the velocity of the water in an axial direc-

tion is the same at each of these points on the second impeller.

Rateau eliminates the end thrust for one condition of running

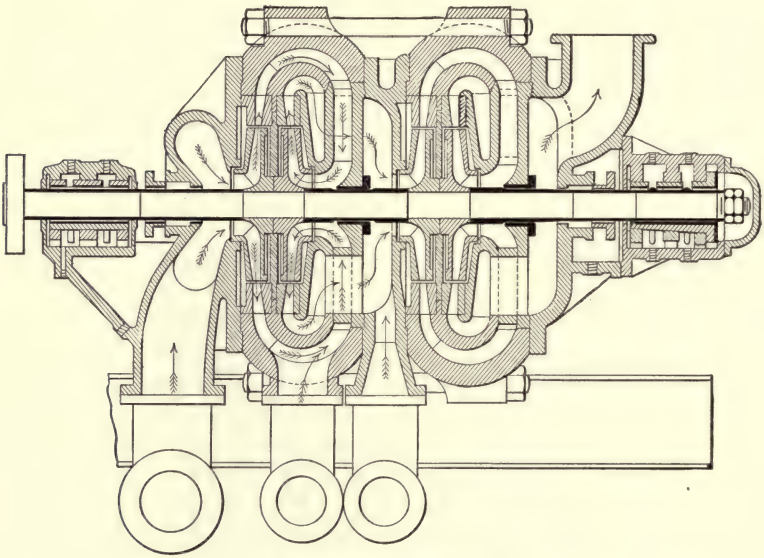

FIG. 445.—Larner Pump.

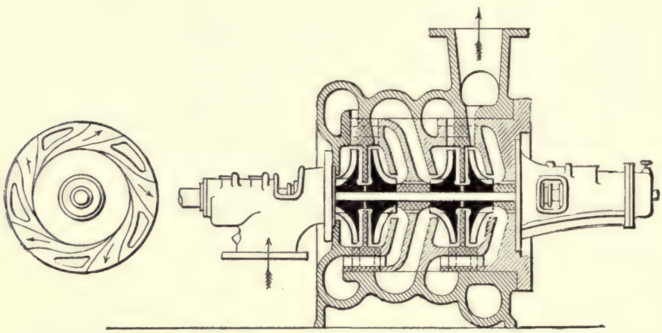

FIG. 446.—Sulzer Pump.

by cutting off one side of the impeller as shown in Fig. 447, and allowing the unbalanced force on the side remaining to balance the pressure at entrance. This means considerable leakage, however, and moreover the wheel for a part of

one side acts as an unencased wheel, which means more friction.

In the three-stage pump of Worthington, Fig. 448, the Jaeger-Worthington method of holding the impeller in a central position is illustrated. In this pump the edge of the diffuser at *A* is cut on each side so that should the impeller move to the right the increased clearance space allows more water to leak out on that side than would be cared for by bushing rings at *B*. As a result the water "backs up" on the right side and the pressure from this forces the impeller to the left. Motion to the left would result in an excess of pressure on that side. To permit this free action the thrust-bearing grooves are made in a sleeve which has a slight play. There is no pressure on the thrust bearing until the sleeve is brought to one end of its travel or the other. In this way the thrust is carried by water on the backs of the impellers and the impeller should take such a position that the excess leakage on one side would care for the thrust.

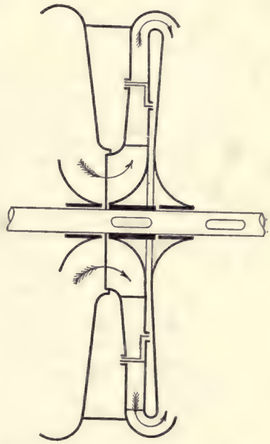

FIG. 447.—Rateau Centrifugal Pump.

In the pump of Weise and Monski the impellers are divided into two groups, one group with the inlets on the right and one set with the inlets on the left. After passing through the first group the water is discharged through a passage in the casing leading to the other end of the pump, where it enters the inlet of the first impeller of the second group. This is the equivalent of having two pumps with their shafts connected, one pump discharging into the other through an outside connection and the inlets to the impellers on one pump turned in the opposite direction from those on the other. Fig. 436 shows such an arrangement.

In the Schwartzkopff type of pump and in the later Sulzer pumps a balancing piston is used at the end of the pump sub-

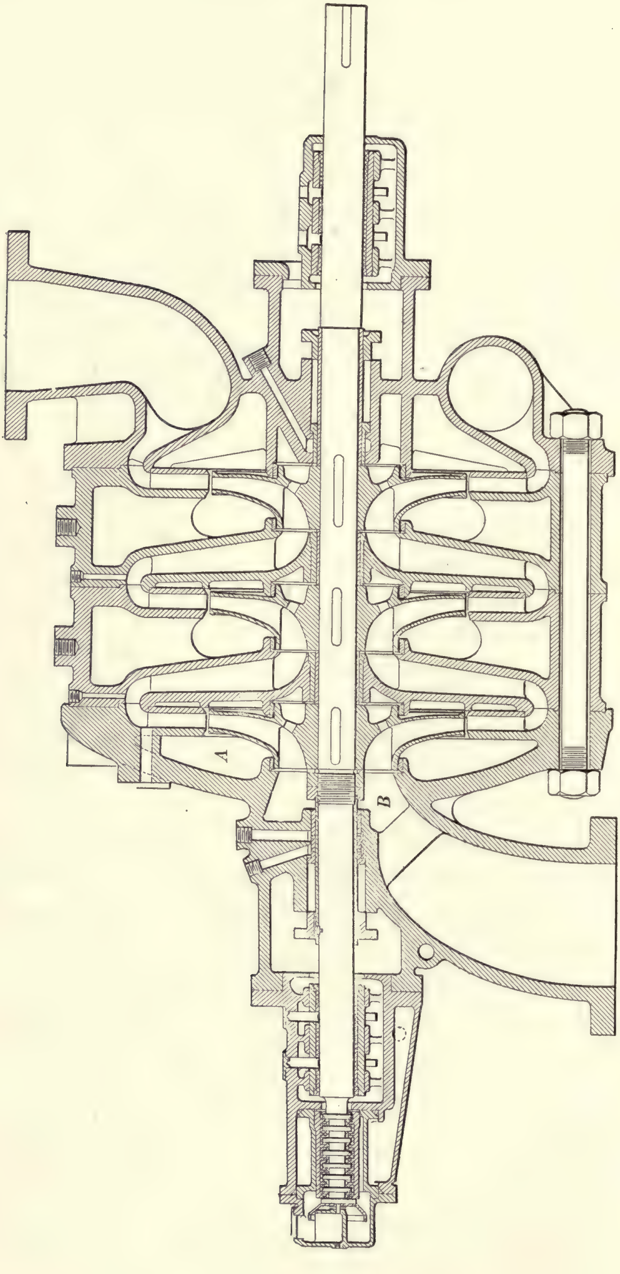

FIG. 448.—Worthington Three-stage Turbine Pump.

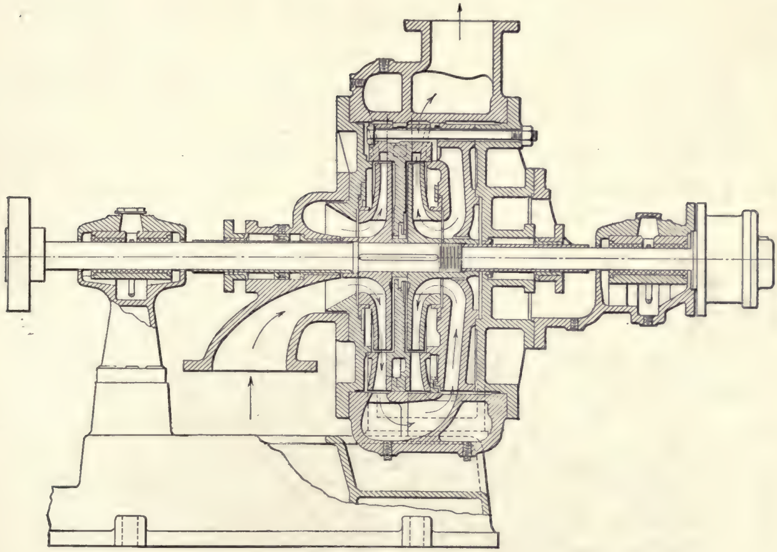

FIG. 449.—Section of a Buffalo Balanced Two-stage Pump.

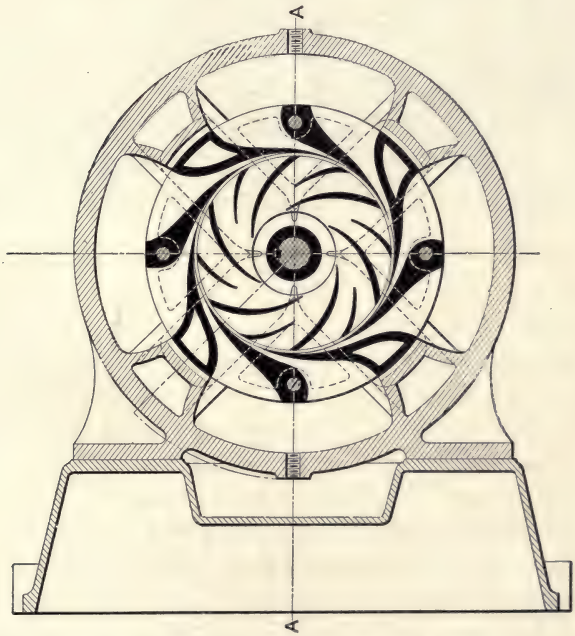

FIG. 450.—Section through Buffalo Pump.

ject to the discharge pressure or a portion of it. The pressure on the piston balances the thrust.

Fig. 449 gives a section through a two-stage Buffalo pump, showing their method of providing for end thrust while the cross-section, Fig. 450, shows the form of passages used to take the water to the second impeller inlet. Fig. 451 shows one of

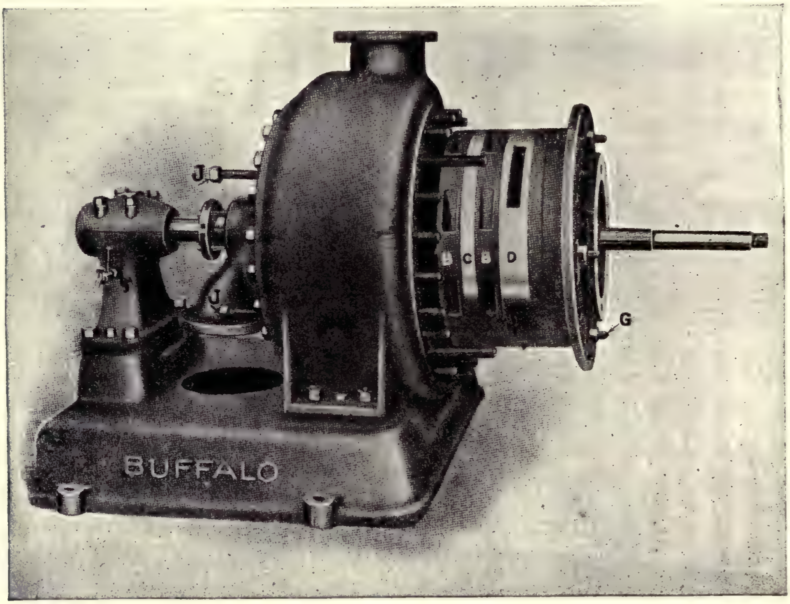

FIG. 451.—Method of Assembling Buffalo Multi-stage Pumps.

these pumps partially dismantled. The outer part of the diffuser vanes *B*,*B* do not fit against the shell, but are bolted by tie bolts *G* to the return chamber *D* and the diaphragm *C*, which reaches to the floating ring around a bushing on the shaft. The return plate *D* and diaphragm *C* have a forced fit into the casing. The bolts *J* are jack bolts used in forcing out the diaphragm *C* and return plate when necessary to examine the interior. In this arrangement of pump it is not necessary to disconnect the suction or discharge pipe when repairing the interior. Passages cast in the casing deliver water from the

left-hand diffuser *B* to the return chamber *D* on the right. These are seen in Figs. 449 and 450.

Fig. 452 shows one of the Buffalo pumps with a vertical

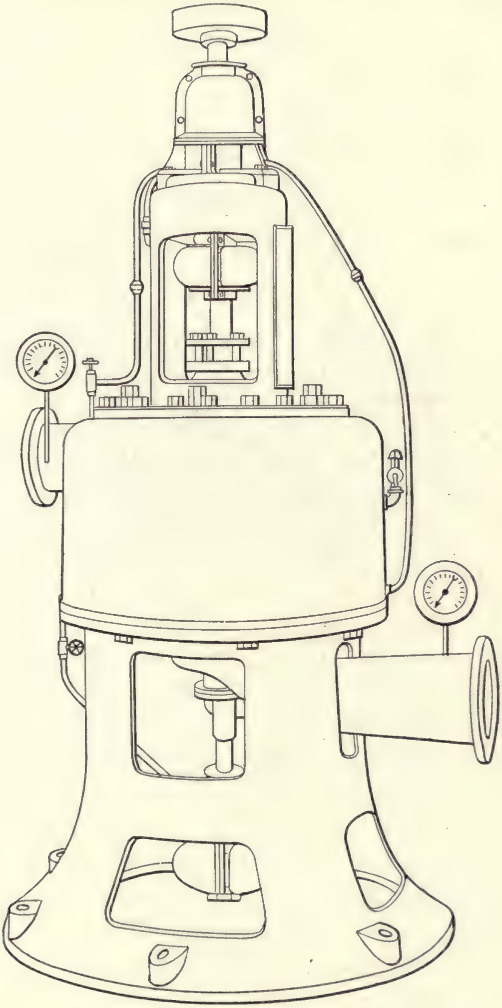

Fig. 452.—Buffalo Vertical Underwriter Fire Pump.

shaft equipped as an underwriter's fire pump. The pump would be driven by a motor through the coupling shown in Fig. 453. This type of coupling is one in which pins on one flange

drive the other flange by means of rubber bushings. Such an arrangement furnishes a yielding medium, so that there is some chance for self alignment when the shafting is deflected in part of the complete machine. In most installations there must be a flexible coupling between the motor and the pump on account of change of alignment at high speeds.

The Allis-Chalmers pumps built for the high-pressure fire service in New York demand mention as another type of multi-stage pump. These pumps, Fig. 454, are built with the passages in the impellers so formed that the outflow is in an axial direction. This form of pump is known as the Gelpcke-Kugel form. These pumps on test gave efficiencies from 72 to 79 per cent.

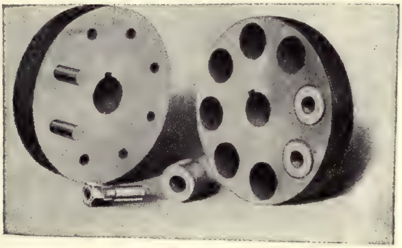

FIG. 453.—Coupling.

There is some objection to the discharge as shown, in that the leakage is apt to be excessive.

Testing Centrifugal Pumps. The centrifugal pump is often designed to run at a fixed speed, being driven by an electric motor. The amount of power used and the efficiency will vary with the quantity of water pumped, and for that reason tests are made to determine the characteristics of the pump.

The characteristics desired are the curves of power, of pressure and of efficiency, plotted against the quantity of water pumped. To obtain these, measurements are made as shown in the diagram, Fig. 455.

The quantity of water may be determined by a Venturi meter, a Pitot tube, a calibrated nozzle or a weir. The head against which the pump is working is determined by two gauges—a suction gauge, and a discharge pressure gauge, or

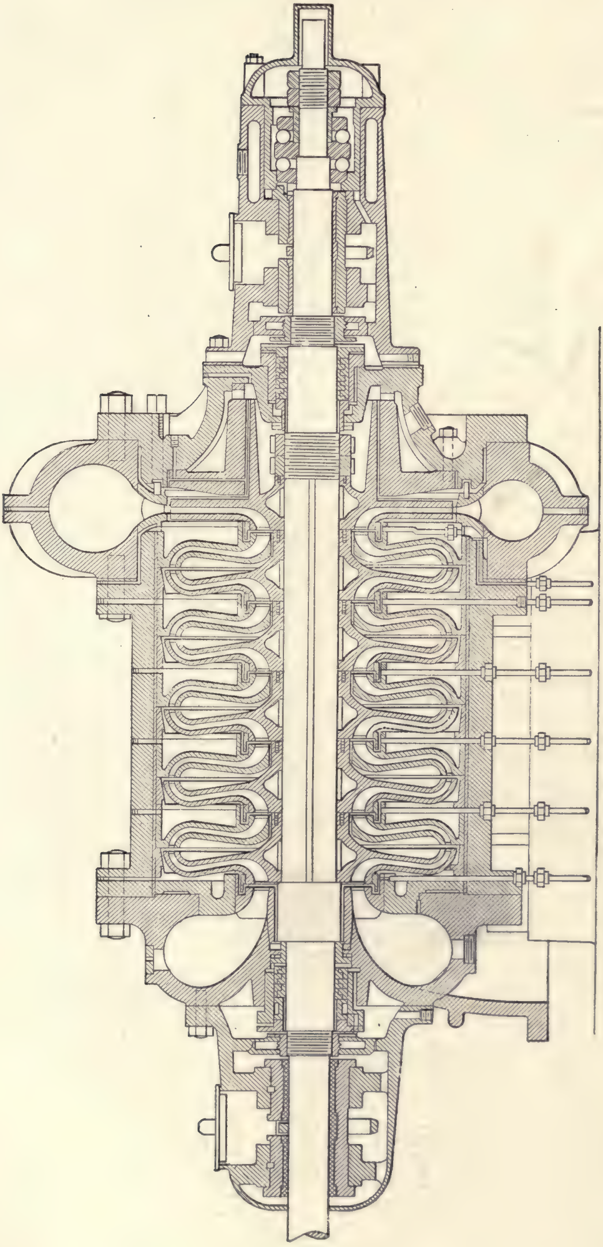

FIG. 454.—Allis-Chalmers Pump for New York.

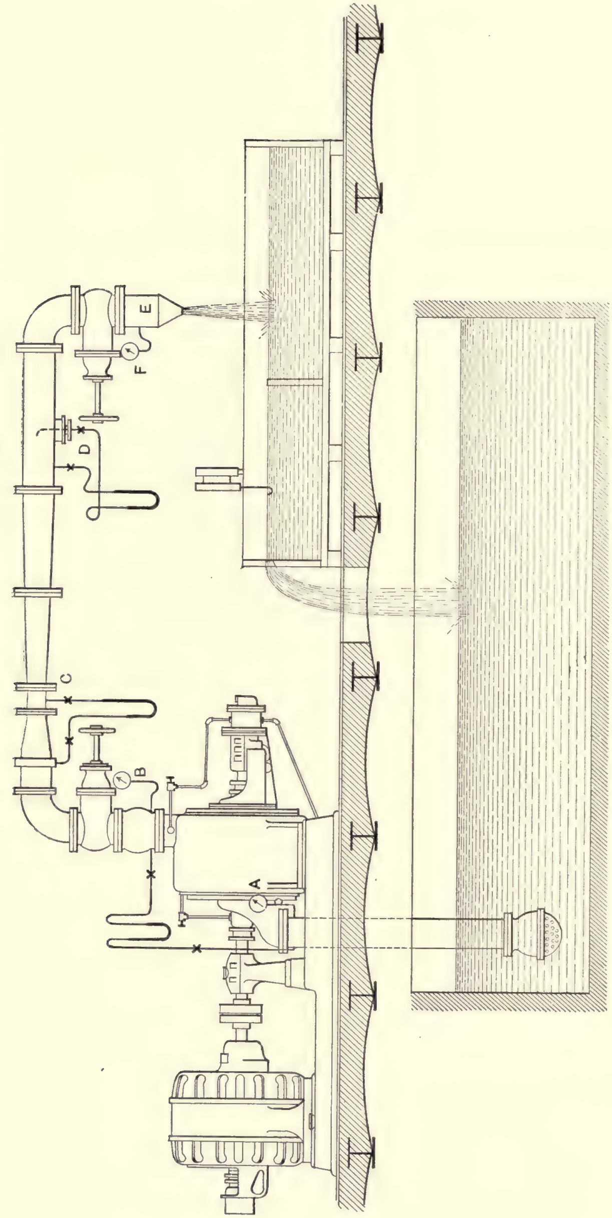

FIG. 455.—Arrangements for Testing Centrifugal Pumps.

the suction may be attached to one end of a mercury U tube while the discharge is connected to the other end. The power input is determined by knowing the power and efficiency of the electric motor, by indicating the engine used to drive the pump or by the use of a dynamometer. The arrangement of the apparatus for such a test is seen in Fig. 455, in which there are several instruments for measuring the water. The suction gauge at A is usually a mercury gauge. The pressure gauge at B may be a Bourdon gauge. The Venturi meter at C, the Pitot tube at D, the nozzle E with its pressure gauge F, or the weir G may be used to determine the water used. The Pitot tube may be applied simply by using a tube flush with the pipe surface for the static pressure tube and sliding the velocity tube across the pipe, making a traverse. With a sharp, small opening in the velocity tube and the tube kept parallel to the axis of the pipe the constant for the tube is unity so that

$$v=\sqrt{2gh},$$

$$q=\int_0^{r_0} 2\pi r dr\sqrt{2gh}=8.02\times 2\pi\int_0^{r_0} r\sqrt{h}\,dr. \quad . \quad . \quad (75)$$

Hence, if $\sqrt{h}$ is multiplied by r and the product is plotted on a straight base for different values of r from 0 to r_0, the area of this curve when multiplied by 16.04π will be the quantity in cubic feet, since all measurements are in feet.

The formula for the Venturi meter, as derived in Merriman's "Hydraulics," is

$$Q=\frac{A_1A_2}{\sqrt{A_1^2-A_2^2}}\sqrt{2g(H_1-H_2)}, \quad . \quad . \quad . \quad (76)$$

where A_1 and A_2 are the areas of the sections of the meter at which the pressures are H_1 and H_2 feet of water.

The formula for the calibrated nozzle is

$$Q=k\sqrt{h},$$

where h is the head shown by the gauge at F measured in feet.

For the rectangular weir the formula of Francis,

$$Q=3.33[b-0.2H]\,[(H+h)^{3/2}-h^{3/2}], \quad . \quad . \quad . \quad . \quad (77)$$

may be used with fair approximation to the correct value. In this H is the head on crest of weir in feet and $h=\frac{v^2}{2g}$ is the head of velocity of approach and b the breadth of the weir. All measurements are in feet. The above is applicable to a weir with end contractions. If contractions are suppressed the term $0.2H$ is omitted.

When the quantity of water is not large a triangular weir may be used with success. This notch is made with sides at right angles and for small quantities the head is much higher than for the rectangular weir, and for that reason there is less error in measuring the acting head. The formula for this weir is

$$Q=0.305H^{5/2}. \qquad (78)$$

The constant has been studied, and although it does vary the limits are not far apart, being 0.291 at 0.7″, 0.306 at $1\frac{1}{2}$″ and 0.303 at 5″. In this formula H is head on crest measured in inches; in all of the above Q is measured in cubic feet.

The head acting not only includes the suction and discharge head, but it should include the velocity head acting at the time.

Having the total head H equal to $h_1+h_2+\frac{v_0{}^2}{2g}$ and the quantity of water Q, the useful power becomes

$$HP_u=\frac{wQH}{550}. \qquad (79)$$

In this $w=$ weight of 1 cubic foot of water, which is determined by actually weighing a known quantity of water, by the use of a hydrometer, or by taking the temperature and referring to a table of weights of water.

The efficiency of the pump is determined when the applied power is found by means of a dynamometer or calibrated electric motor.

$$\text{Eff.}=\frac{HP_u}{HP_a}. \qquad (80)$$

The results of a test are usually plotted with the quantity of discharge as abscissæ using head, efficiency, and applied

power as ordinates of three separate curves. Fig. 456 shows such curves from a Worthington 8″ volute pump. In this pump the head remains constant for a considerable time as the discharge valve is opened, so that more water is discharged until 1800 gallons per minute is reached, when the head begins to drop rapidly, accompanied by a decrease of efficiency and a slight increase of power. It is to be noted that the increase of power begins to fall off at high discharges, due to the decrease

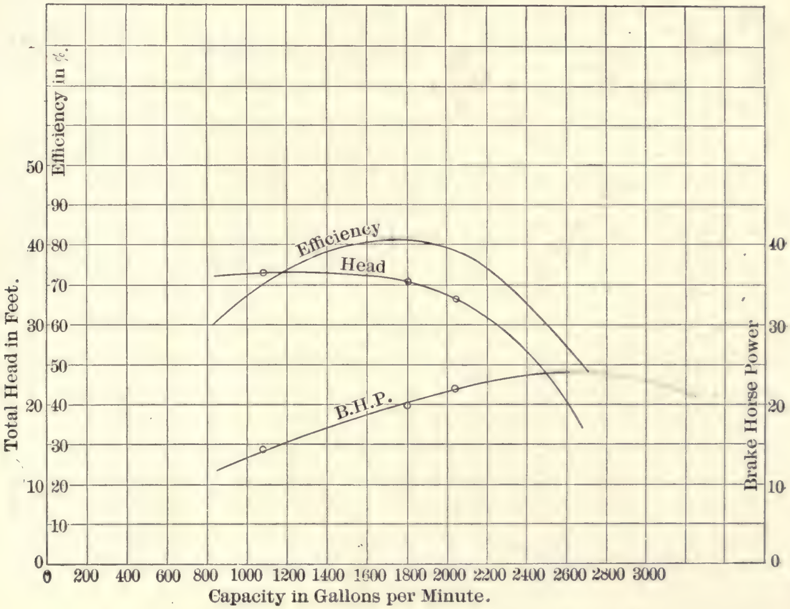

FIG. 456.—Test Curves from an 8″ Worthington Volute Pump.

of head. The head does not increase as the discharge valve is closed and the power decreases, due to the decrease in quantity. The efficiency has a maximum point because when the quantity changes from the designed quantity, the water passes through the wheel with impact. This may be seen when it is remembered that the speed of the pump is fixed and the velocities in the channels are fixed by the equation

$$v = \frac{Q}{A}. \qquad \qquad (81)$$

In Fig. 457, from a Worthington class B volute pump, the head is seen to be changed more rapidly than in the preceding figure. The efficiency reaches a higher value while the brake horse power of the motor, which is delivered to the pump, actually decreases as the quantity passes beyond 4000 gallons per minute; this is associated with a head of 44 feet. The power curve is quite characteristic of a centrifugal pump, as

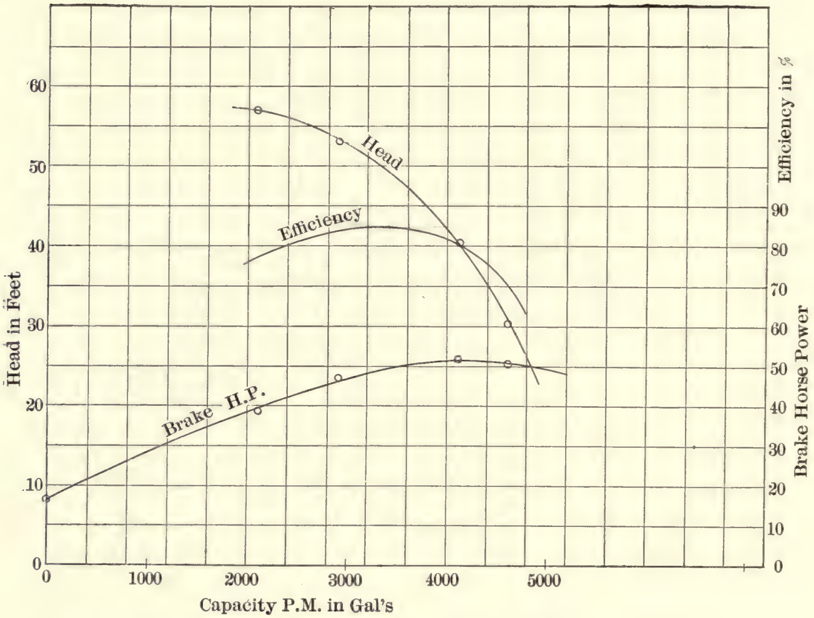

FIG. 457.—Test Curves of a 10″ Worthington Class B Volute Pump.

it shows how, with a proper design, these is no danger of overloading the motor if the water is shut off (no discharge) or if the discharge pipe should break (maximum discharge).

Fig. 458 gives the characteristic curve from a six-stage turbine pump used in the Brooklyn High Pressure Station. In this figure the entire range from zero discharge to maximum discharge under no head is shown. The horse power is seen to have a maximum value at 840 H.P. and the maximum efficiency is seen to reach the value of 76 per cent. The

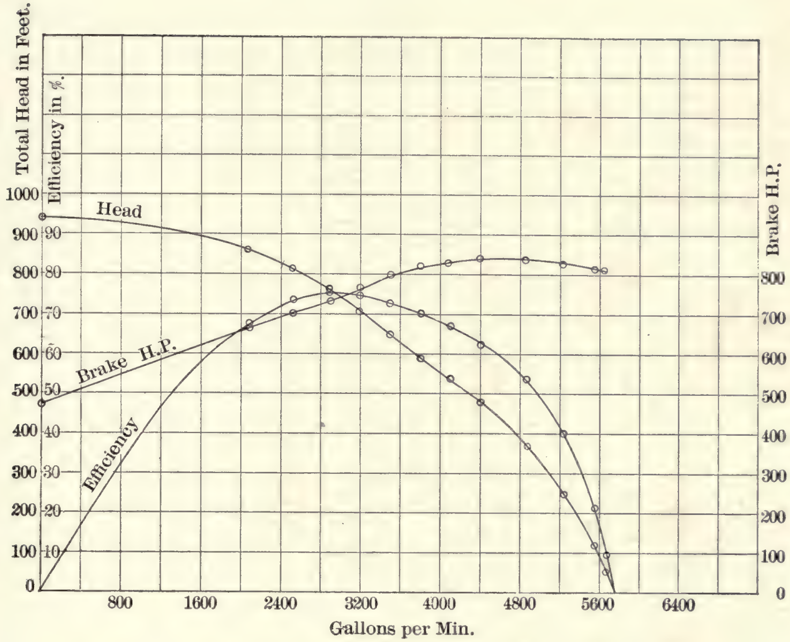

FIG. 458.—Test Curves of 10″ Six-stage Worthington Turbine Pump.

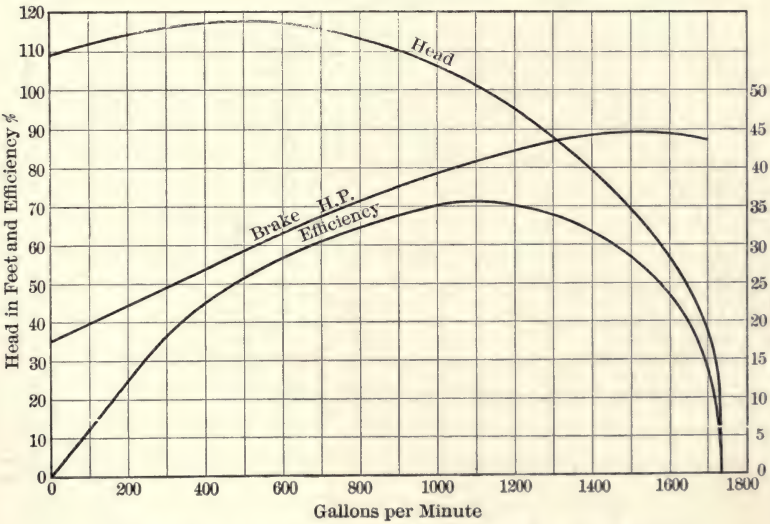

FIG. 459.—Test Curves of an Alberger Single-stage Pump.

high efficiencies on these three diagrams are important to notice. The characteristic curves from a 10″ single-stage turbine pump of the Alberger design is seen in Fig. 459. In this the complete range of the pump is given. Fig. 460 gives the excellent results of a pump built by the I. P. Morris Co.

These curves show how the centrifugal pump is suited to conditions under which the head would vary or where the quantity would vary. In the case of emptying drydocks where the head changes as the dock is emptied, a volute pump would be used. At the start the quantity would be great at a very small head, while as the head increases the quantity decreases, and in some cases the power required increases up to a certain point, after which there is a decrease of power and quantity as the head increases.

In the case of a fire pump (Fig. 458) the variation of pressure with the change in quantity is evident; with 4 400-gallon streams the head is 900 feet, while with 8 such streams the head is reduced to 705 feet. When 12 400-gallon streams or 19 250-gallon streams are used, the pressure is reduced to 380 feet. In none of these cases is there danger of overloading the proper-sized motor. In selecting a pump it is better to select one in which the characteristic curve of head gives a maximum value at zero discharge, as there will be no difficulty in obtaining the head for which the pump was designed. If it is desired to keep the head constant at the various discharges, the speed of the pump will have to be increased, and with this the power of the motor. A speed characteristic could be drawn for any variation of head from the head curve at a given speed by remembering that the head varies as the square of the speed.

In order to classify pumps a similar method to that used in turbine classification may be employed.

Specific Speed. One of the powerful aids in the classification of turbines for the purpose of design so that a given design may be used for any turbine having a given characteristic, is the specific speed. This is the speed of a turbine of design similar to a given turbine but of different scale, such that under a unit head it would develop one unit of power. A

similar characteristic may be used for a centrifugal pump; here the specific speed of a pump will be that speed at which

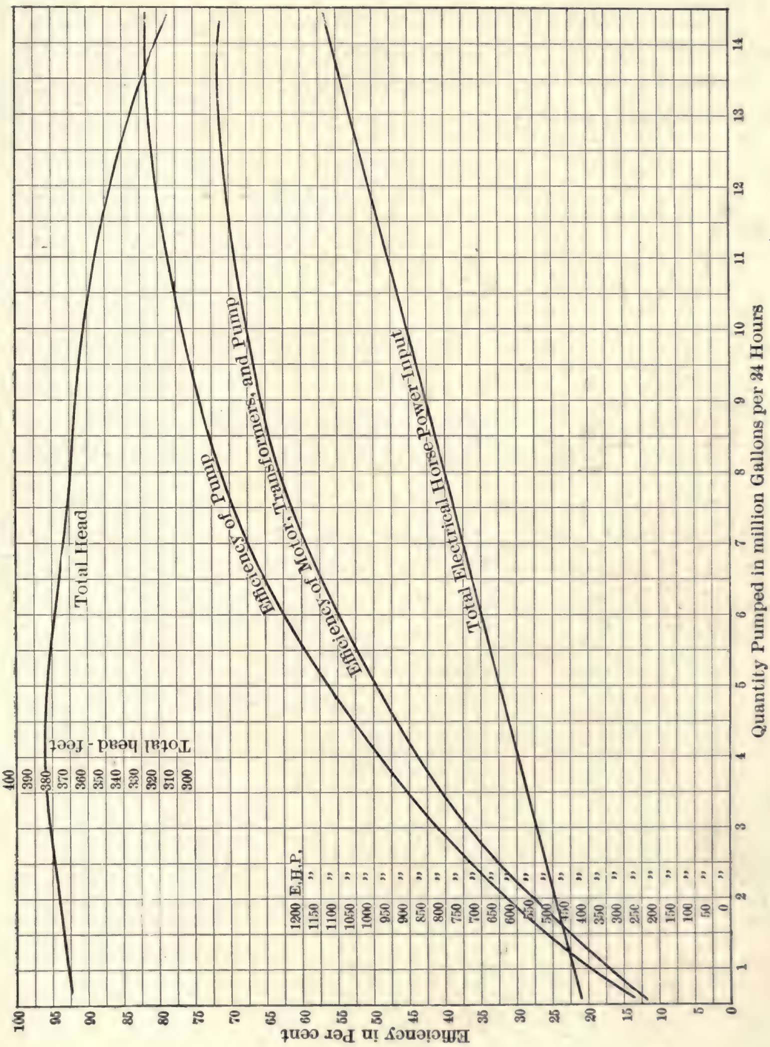

FIG. 460.—Test of an I. P. Morris Co. Pump for Duluth.

a pump of similar design, but to a different scale, will pump one unit volume of water, through a total head of unity. To derive the formula for the specific speed of a centrifugal pump a

method will be used similar to that used by Professor L. F. Moody for the specific speed of turbines as given in Zeitschrift f. d. gesampte Turbinenwesen, Sept. 10, 1909.

Suppose that the quantity of water lifted by a pump running at N_a revolutions is Q_a and the total head is kH. If this wheel is now run under a total head of k_1H_1 the speed will be changed so that

$$N_1=N_a\sqrt{\frac{k_1H_1}{kH}}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (82)$$

because

$$N\propto V\propto \sqrt{H}.$$

The quantity is equal to the area multiplied by the velocity, and hence

$$Q_1=Q_a\sqrt{\frac{k_1H_1}{kH}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (83)$$

If now the scale of the pump be changed so that the quantity of water is changed to Q_1' the diameter being changed from D_a to D_1'.

Now

$$Q\propto A\propto D^2,$$

hence

$$Q_1'=Q_1\frac{D_1'^2}{D_a^2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

When D_a is changed to D_1' the speed in revolutions per minute will be changed in the inverse manner because the velocity remains the same as the head remains constant, hence

$$N_1'=\frac{D_a}{D_1'}N_1, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

but

$$\frac{D_a}{D_1'}=\sqrt{\frac{Q_1}{Q_1'}}=\left(\sqrt{\frac{Q_a\sqrt{\frac{k_1H_1}{kH}}}{Q_1'}}\right),$$

and

$$N_1 = N_a\sqrt{\frac{k_1H_1}{kH}}.$$

Therefore

$$N_1' = N_a\sqrt{\frac{Q_{a'}}{Q_1'}}\left(\frac{k_1H_1}{kH}\right)^{3/4}. \quad . \quad . \quad . \quad . \quad .$$

If N_1' is to be the specific speed, the total head K_1H_1 will be unity and the quantity Q_1' will be unity.

Hence

$$N_1' = N_s = N_a\frac{\sqrt{Q}}{\sqrt[4]{(kH)^3}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

In this N is taken as R.P.M.; Q, cubic feet per section per second; kH, feet head.

This could be derived from the expression for the specific speed of a turbine by substituting QH for HP in the formula (for turbines),

$$N_s = N_a\frac{\sqrt{HP}}{(H)^{5/4}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

This formula can be used in either the English or French system of units. In the English system Q is in cubic feet per second and H in feet, while in the French system Q is in cubic meters per second and H is in meters. The value N_s in the French units may be changed to N_s for the English system by multiplying by $\frac{1}{2.438}$.

The value of the specific speed for a series of different pumps has been plotted in Fig. 461 with the efficiency of the pump as ordinates, and from it the specific speed for the best efficiency may be found.

The specific speed of a pump is the same for all similar pumps, and for a given quantity, head and number of revolutions the specific speed may be found. This immediately tells the designer the class to which the pump belongs and what assumptions are to be made. Should the specific speed be too high, the pump could be made of several single stages in parallel,

as the quantity Q in the formula refers to the quantity for one impeller and the head H to the head for that impeller. For a multistage pump the quantity Q is the quantity for the whole pump while H is that for one stage. Should N_s in a desired plant be too small for high efficiency, the use of a multistage pump would require a higher specific speed and so give a better efficiency.

Designing Centrifugal Pumps

To illustrate the methods of design, applying the principles of this chapter, a pump will be designed to lift 1000 gallons per minute through 350 feet of pipe against a static head of 300 feet, using a motor at 750 R.P.M. or 1000 R.P.M. It is assumed that these are the only speeds possible. The calculations are made with a slide rule.

1. *Specific Speed and Number of Stages.*

$$1000 \text{ gals. per min.} = \frac{1000}{60 \times 7.48} = 2.23 \text{ cu.ft. per sec.}$$

From (87),

$$N_s = N\frac{\sqrt{Q}}{H^{\frac{3}{4}}}.$$

From the curve, Fig. 461, 75 appears to be a good specific speed, hence

$$N_s = 75 = \frac{750\sqrt{2.23}}{H^{\frac{3}{4}}},$$

$$H = 36.6 \text{ ft.}$$

This is a very small head for one stage, and consequently a smaller specific speed might be used or a higher actual speed. Decreasing the specific speed means a smaller efficiency, but one which is not reduced very much if proper design is used, using 60 for N_s and the higher speed of 1000 R.P.M.

$$H^{\frac{3}{4}} = \frac{1000}{60}\sqrt{2.23} = 25.0.$$

$$H = 73.0.$$

This is a good head to use, requiring 3 or 4 stages. Using 3 stages and considering the losses in head to be 5%,

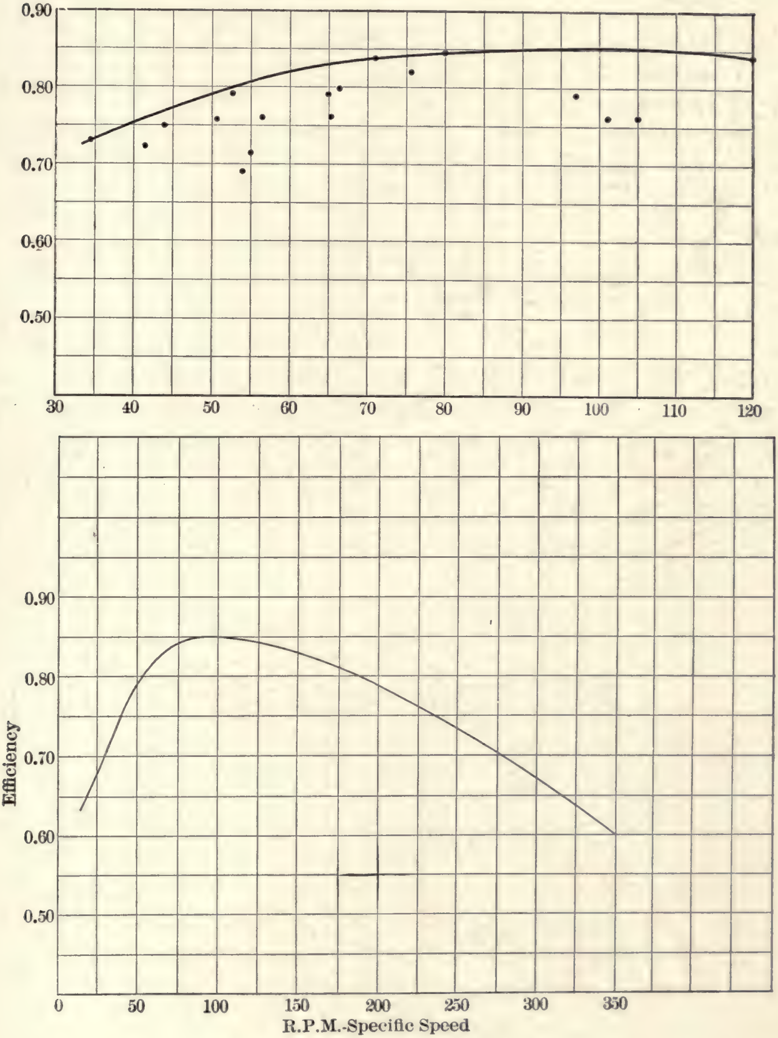

FIG. 461.—Specific Speed Curves.

$$N_s = \frac{1000\sqrt{2.23}}{\left(\frac{315}{3}\right)^{\frac{3}{4}}} = 45.5.$$

This is not as good a specific speed as could be obtained if we desired to use another stage. In that case

$$N_s = \frac{1000\sqrt{2.23}}{\left(\frac{315}{4}\right)^{\frac{3}{4}}} = 58.4.$$

Could the speed of the pump be changed by gears, the actual speed, for $N_s = 75$, and 3 stages would be,

$$N = \frac{N_s H^{\frac{3}{4}}}{\sqrt{Q}} = \frac{75 \times \left(\frac{315}{3}\right)^{\frac{3}{4}}}{\sqrt{2.23}} = 1650.$$

This is the speed which should be used with this pump for best conditions but with given data either $N_s = 58$ or $N_s = 46$ will be taken. To keep down the number of impellers the second of these will be employed. From the curve of Fig. 461, the efficiency is assumed as 78%.

2. *Size of Suction and Discharge Pipe.* With a short pipe

$$v_s = 8.02\sqrt{\frac{300}{50}} \text{ approx. from (61)}$$

$$= 19.6 \text{ ft. per sec.}$$

$$A_s = \frac{2.23}{19.6} \times 144 = 16.38 \text{ sq.ins.}$$

$$\therefore d_s = 4\tfrac{9}{16} \text{ in.}$$

Use $5'' = d_s$.

$$A_s = 19.63,$$

$$v_s = \frac{2.23 \times 144}{19.63} = 16.35 \text{ ft. per sec.}$$

Loss assuming length to be 20 ft.,

$$\zeta_s \frac{v_s^2}{2g} = \left[\frac{1}{2} + \frac{0.02 \times 20}{\frac{5}{12}}\right] \frac{(16.35)^2}{64.32} = 6.2 \text{ ft.}$$

If discharge pipe is of the same size the loss would be

$$\zeta_0 \frac{v_0^2}{2g} = \left(\frac{0.02 \times 330}{\frac{5}{12}} \right) \frac{(16.35)^2}{64.32} = 65.7 \text{ ft.}$$

This loss is over 20% of net lift and hence a larger pipe will be taken for the discharge, say, 8 inches.

$$A_d = 50.26 \text{ sq.ins.}$$

$$v_d = \frac{2.23 \times 144}{50.26} = 6.38 \text{ ft. per sec.,}$$

$$\zeta_0 \frac{v_0^2}{2g} = \frac{0.02 \times 330}{\frac{8}{12}} \frac{(6.38)^2}{64.32} = 6.25 \text{ ft.}$$

This is a permissible loss.

If a 10-inch pipe is tried:

$$\zeta_0 \frac{v_0^2}{2g} = 2.1 \text{ ft.}$$

The saving in the use of the larger pipe is

$$\frac{(6.2 - 2.1)\, 2.23 \times 62.5}{550} = 1.04 \text{ H.P.}$$

If the pump has an efficiency, from the curve at a specific speed of 46, of 80%, if the motor has an efficiency of 90%, if the pump runs 4000 hours per year and if power cost 1 cent per K.W. hour, the saving per year is

$$\frac{1.04}{0.80 \times 0.90} \times 0.746 \times 4000 \times 0.01 = \$43.20.$$

The additional cost of pipe is

$$\left[W \frac{\pi(d - d)t \times l}{2000} \times \text{cost per ton} \right] [1 + \text{installing factor}],$$

$$\left[\frac{0.28\pi(10 - 8) \times 330 \times 12}{2000}\ \$25.00 \right] [2] = \$174.60.$$

Now the interest, depreciation, taxes, and insurance on this pipe line will be taken at 9%, so that the yearly charge will be

$$0.09\times\$174.60=\$15.71.$$

It is seen that the saving per year from the use of a 10-inch pipe over an 8-inch one is $43.20, while the increased cost is $15.71 per year. The 10-inch pipe will be used and connected to the volute casing by a reducer from 8 inches. A larger pipe should be tried in the same manner. If the saving in power is greater than the increased cost on investment, the larger pipe should be used.

To aid in solving problems of this kind the table on page 607, taken from one of the bulletins of H. R. Worthington, is given.

In this table the amount of water usually carried in a pipe of a certain size is given together with the loss in head in 1000 feet of straight pipe. Since the drop in head varies with the length of pipe this can be used for various lengths. For large pipes the quantities are given in million gallons per twenty-four hours.

The loss in the wheel is assumed to be $0.3\frac{v_s^2}{2g}=1.2.$

The total head is now

$$KH=[1+l+r]H=\left[1+\left(\frac{6.2+1.2+2.1}{300}\right)+\left(\frac{0.25}{300}\right)\right]300$$

$$=310.0 \text{ ft. instead of } 315 \text{ assumed.}$$

3. *Probable Brake H.P. of Motor and Size of Shaft.*

$$\text{B.H.P.}=\frac{2.23\times62.5\times310.0}{550\times0.78}=100 \text{ B.H.P.}$$

The shaft to transmit this power at 1000 will have a diameter given by the equation below if no bending be considered:

$$d=\sqrt[3]{\frac{321{,}000 \text{ H.P.}}{NS}}.$$

$$d=\sqrt[3]{\frac{321{,}000\times100}{1000\times10{,}000}}=1.48.$$

TABLE SHOWING LOSS OF HEAD IN FEET PER 1000 FEET OF STRAIGHT PIPE

Size Pipe Ins.																		Size Pipe Ins.
1	Gals. per min.	3	4	5	6	8	10	12	14	16	18	20	25	30	35	40	Gals. per min.	1
	Friction head	12.6	21.4	32.5	45.5	78	117	164	220	280	350	420	640	890	1190	1520	Friction head	
1½	Gals. per min.	5	10	15	20	25	30	35	40	45	50	60	70	80	90	100	Gals. per min.	1½
	Friction head	3.9	14.3	30	52	78	110	147	188	232	284	396	530	680	840	1020	Friction head	
2	Gals. per min.	25	30	35	40	45	50	60	70	80	90	100	125	150	175	200	Gals. per min.	2
	Friction head	27.3	38.4	51	66	82	99	139	184	237	294	358	540	760	1015	1290	Friction head	
3	Gals. per min.	50	60	70	80	90	100	120	140	160	180	200	225	250	300	350	Gals. per min.	3
	Friction head	13.8	19.2	25.7	32.8	40.8	49.6	70	92	118	148	178	225	271	380	504	Friction head	
4	Gals. per min.	100	120	140	160	180	200	240	280	320	360	400	450	500	550	600	Gals. per min.	4
	Friction head	12.2	17.1	22.8	29.1	36.1	44	62	82	105	131	160	198	240	287	337	Friction head	
5	Gals. per min.	160	180	200	240	280	320	360	400	450	500	550	600	700	800	900	Gals. per min.	5
	Friction head	9.8	12.2	14.8	20.8	27.7	35.4	45	54	67	81	96	113	151	194	240	Friction head	
6	Gals. per min.	200	230	260	300	340	380	440	500	600	700	800	900	1000	1200	1400	Gals. per min.	6
	Friction head	6.1	7.9	10	13.5	16.0	20.0	26.1	33.3	46.4	61	79	100	122	167	222	Friction head	
8	Gals. per min.	350	400	450	500	600	700	800	900	1000	1200	1400	1600	1800	2000	2200	Gals. per min.	8
	Friction head	4.2	5.5	6.8	8.2	11.5	15.1	19.5	24.5	29.9	40.0	54	71	89	108	128	Friction head	
10	Gals. per min.	700	750	800	900	1000	1200	1400	1600	1800	2000	2200	2400	2600	2800	3000	Gals. per min.	10
	Friction head	5.1	5.9	6.6	8.3	10.0	13.8	18.4	23.6	29.9	36.0	43.5	51	58.5	66	77.2	Friction head	
12	Gals. per min.	1400	1500	1600	1700	1800	2000	2200	2400	2600	2800	3000	3500	4000	4500	5000	Gals. per min.	12
	Friction head	7.6	8.8	9.7	10.9	12.2	15.0	18.1	21.0	24.1	27.0	30.9	38.1	53.7	67.5	81.4	Friction head	
16	Gals. per min.	3000	3200	3400	3600	3800	4000	4500	5000	5500	6000	6500	7000	8000	9000	10000	Gals. per min.	16
	Friction head	7.8	8.8	9.9	11.0	12.1	13.2	16.6	20.1	24.0	28.0	32.8	36.6	48.0	60.0	72.0	Friction head	
20	Gals. per min.	6000	6250	6500	7000	7500	8000	8500	9000	9500	10000	11000	12000	13000	14000	15000	Gals. per min.	20
	Friction head	9.5	10.2	11.2	12.5	14.3	16.1	17.9	20.0	22.2	24.3	29.0	34.2	39.5	44.6	51.3	Friction head	
24	Mil. gals. 24 hrs.	11	12	13	14	15	16	17	18	19	20	22	24	26	28	30	Mil. gals. 24 hrs.	24
	Friction head	6.1	7.1	8.3	9.5	10.8	12.2	13.6	15.2	16.7	18.3	21.9	25.9	29.9	34.2	39.0	Friction head	
30	Mil. gals. 24 hrs.	13	14	15	16	17	18	19	20	22	24	26	28	30	35	40	Mil. gals. 24 hrs.	30
	Friction head	2.8	3.2	3.6	4.1	4.6	5.1	5.6	6.2	7.4	8.7	10.1	11.6	13.2	17.5	22.4	Friction head	
36	Mil. gals. 24 hrs.	20	22	24	26	28	30	32	34	36	38	40	45	50	55	60	Mil. gals. 24 hrs.	36
	Friction head	2.5	3.0	3.5	4.1	4.7	5.4	6.1	6.8	7.6	8.4	9.2	11.4	13.9	16.6	19.6	Friction head	
42	Mil. gals. 24 hrs.	34	36	38	40	42	44	46	48	50	55	60	65	70	75	80	Mil. gals. 24 hrs.	42
	Friction head	3.2	3.6	4	4.35	4.8	5.2	5.6	6.1	6.6	7.8	9.2	10.7	12.2	13.9	15.7	Friction head	
48	Mil. gals. 24 hrs.	42	44	46	48	50	55	60	65	70	75	80	85	90	95	100	Mil. gals. 24 hrs.	48
	Friction head	2.5	2.7	3.0	3.2	3.5	4.1	4.8	5.6	6.4	7.3	8.2	9.2	10.2	11.3	12.4	Friction head	
54	Mil. gals. 24 hrs.	46	48	50	55	60	65	70	75	80	85	90	95	100	110	120	Mil. gals. 24 hrs.	54
	Friction head	1.7	1.8	2.0	2.3	2.7	3.2	3.6	4.1	4.6	5.2	5.8	6.4	7.0	8.4	9.4	Friction head	

On account of bending to be considered later as well as the effect of critical speed, the diameter will be taken as

$$d_a = 1\tfrac{1}{3}d = 2 \text{ ins.}$$

4. *Angular Arrangement of Vanes and Areas at Entrance and Discharge.* From Newmann's formulæ (36), (46),

$$w_2 = x\sqrt{gKH},$$

$$u_r = \frac{\sqrt{gKH}}{x} \cot \beta.$$

The area of outlet measured on the circumference is

$$A_r = \left(\pi D_2 - \frac{nt'}{\cos \alpha_2}\right) b_2, \quad . \quad . \quad . \quad . \quad . \quad (89)$$

where n = number of vanes;
t' = thickness of vanes;
b_2 = breadth of opening.

For a given diameter, the breadth b_2 is given by the formula,

$$b_2 = \frac{Q_2}{\left(\pi D_2 - \frac{nt'}{\cos \alpha_2}\right) u_r} = \frac{Q_2}{\left(\pi D_2 - \frac{nt'}{\cos \alpha_2}\right)\sqrt{gKH}} \frac{x}{\cot \beta}.$$

If now H is small, b_2 may be of proper size with a small value of x and a large value of cot β, but when H is large x should be as large as possible and cot β as small as possible, so as to make b_2 of proper value. Now

$$w_2 = x\sqrt{gKH}.$$

From this it is seen that it is better to keep x as small as possible consistent with other demands in order to make w_2 small, because the friction of water against the impeller depends on this term.

On examining Fig. 413 it is seen in the first place that when $\beta=90°$, $x=1$ for all values of α_2 and also that with values of β near 90° there is little change in x. Consequently for high heads β should be made as near 90° as possible. It is true that with a negative value of α_2 x is larger for smaller values of β, but cot β increases so much that $\frac{x}{\cot \beta}$ might not be large. When the pump has a low head opposing flow, x may be smaller and β smaller, giving cot β a larger value. For high-pressure stages β may be made 85°, while with low-pressure stages β may be 40 to 50°. The value of α_2 is usually negative, 30 to 60° being used. By using positive values of α_2, the speed of the pump w_2 may be decreased, giving what may be called bending back. If α_2 is made 0°, all values of β give $x=1$, hence $w_2=\sqrt{gKH}$. For pumps in which there is no diffuser β is made almost 0°, so that u may be small, giving p_d as large a value as possible, since there is no good method of changing velocity into pressure in a volute pump. This will mean that α_2 has a large negative value, say, $-70°$.

Using in the problem, which is a high-pressure stage problem, $\alpha_2=-60°$ and $\beta=85°$, $x=1.073$, $KH=104$ feet, the following results:

$$w_2=1.073\sqrt{32.16\times 104}=62.0;$$

$$N=1000 \text{ per min.}=16.7 \text{ per sec.};$$

$$r_2=\frac{62}{\frac{100}{6}\times 2\pi}=0.59 \text{ ft.}=7 \text{ ins.}$$

From (47) and (48), using Fig. 414, $\lambda=0.0038$,

$$u_r=\frac{\sqrt{\lambda 2gKH}}{x}=\frac{\sqrt{0.0038\times 64.32\times 104}}{1.073}=4.7;$$

$$u=\frac{\sqrt{gKH}}{x}\sqrt{1+2\lambda}=\frac{\sqrt{32.16\times 104}}{1.073}\sqrt{1.0078}=53.9.$$

As a check: $u \cos \beta = u_r = 53.9 \times 0.08716 = 4.7$,

$$b_2 = \frac{Q}{u_r\left(\pi D_2 - \dfrac{nt'}{\cos \alpha_2}\right)} = \frac{2.23 \times 144}{4.7\left(\pi \times 14 - \dfrac{6 \times \frac{1}{16}}{\frac{1}{2}}\right)} = 1.58 \text{ ins.,}$$

assuming six vanes $\frac{1}{16}$ inch thick at point.

To find the data for entrance certain assumptions must be made. $\dfrac{r_2}{r_1}$ varies from $1\frac{1}{2}$ to $2\frac{1}{2}$. c_r may be made equal or nearly equal to u_r. It is better to have $u_r < c_r$, as this means that the velocity at discharge is small.

Let $c_r = 1.75 u_r = 1.75 \times 4.7 = 8.22$.

This is the velocity c since $\alpha = 0°$. If the water entering the impeller have this same velocity the area between the hub and the shrouding or wall of the impeller must be given by the equation,

$$_h = A = \frac{Q}{c} = \frac{2.23 \times 144}{8.22} = 39.0 \text{ sq.ins.}$$

5. *Diameters of Impeller.* The hub of the impeller may be taken to be $\frac{3}{4}$ inch thick, giving the diameter,

$$D_h = 2'' + 2 \times \tfrac{3}{4} = 3\tfrac{1}{2};$$

$$A_h = 9.6 \text{ sq.ins.}$$

The area of circle at inner end of the shrouding will be

$$A_s = 9.6 + 39.0 = 48.6;$$

$$D_s = 7.82 = 7\tfrac{7}{8}.$$

Using 8 inches for the diameter at the edge of the vane of the impeller gives

$$\frac{r_2}{r_1} = \frac{14}{8} = 1.75, \text{ a workable ratio.}$$

Now $c_r = 1.75 u_r = 1.75 \times 4.7 = 8.22$;

$$w_1 = \frac{w_2}{1.75} = \frac{4}{7}\,62 = 35.4;$$

$$d_1 = 8 \text{ ins.};$$

$$\tan \alpha_1 = \frac{w_1}{c_r} = \frac{35.4}{8.22} = 4.31;$$

$$\alpha_1 = 76^\circ - 55';$$

$$b_1 = \frac{Q}{c_r\left(\pi D_1 - \dfrac{nt'}{\cos \alpha_1}\right)} = \frac{2.23 \times 144}{8.22\left(\pi \times 8 - \dfrac{6 \times \frac{1}{16}}{0.23}\right)} = 1.66 \text{ ins.}$$

6. *Alternate Graphical Method.* The above dimensions may be found in another manner. From Eq. (30),

$$u = \sqrt{\frac{2gH}{2\dfrac{r_2}{r_1}\dfrac{A_{ex}}{A_1}\sin \alpha_1 \sin \beta - \Sigma\zeta - \left(\dfrac{A_{ex}}{A_d}\right)^2}}.$$

The ratios $\dfrac{r_2}{r_1}$, $\dfrac{A_{ex}}{A_1}$, $\dfrac{A_{ex}}{A_d}$, and the quantities α_1, β, and $\Sigma\zeta$ are assumed from practice. Let

$$\frac{r_2}{r_1} = 1.75, \quad \frac{A_{ex}}{A_1} = \frac{5}{8}, \quad \frac{A_{ex}}{A_d} = \frac{1}{5}, \quad \beta = 85^\circ, \quad \alpha_1 = -75^\circ,$$

and ζ for the impeller and for the diffuser each 0.15. Then

$$u = \sqrt{\frac{64.32 \times 100}{2 \times 1.75 \times \frac{5}{8} \times 0.996 \times 0.966 - 0.3 - \frac{1}{25}}} = 60.4;$$

$$c_1 = \frac{A_{ex} u}{A_1} = 60.4 \times \tfrac{5}{8} = 37.7.$$

In Fig. 462 ABC is taken of such a length that AB repre-

sents r_1 and AC, $r_2\left(\frac{AB}{AC}=\frac{1}{1.75}\right)$ and BD is drawn perpendicular to AB, and BE is drawn, making an angle of 75° ($\alpha_1=75$) with AB. Lay off BE equal to 37.7 feet per second. A perpendicular

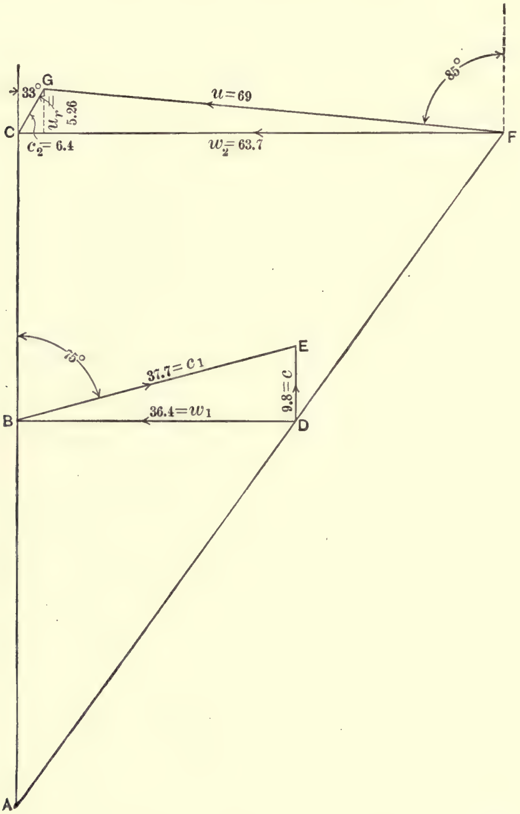

FIG. 462.—Graphical Design.

ED from E on BD will fix the velocities $w_1=DB=36.4$ and $c=ED=9.8$.

A line is drawn from A through D cutting a perpendicular to AC from C in the point F. FC is equal to w_2. From the figure this equals 63.7, FG is laid off at 85° to a line parallel to AC and is made equal to 69 feet per second. This gives $CG=c_2=6.2$, $\alpha_2=-33°$.

From the figure $u_r = 5.26$, $c_r = c = 9.8$;

$$D_2 = \frac{63.7 \times 12}{\frac{100}{6} \times \pi} = 14.6 \text{ ins.};$$

$$D_1 = \frac{37.7 \times 12}{\frac{100}{6} \times \pi} = 8.65 \text{ ins.};$$

$$b_2 = \frac{2.23 \times 144}{5.26\left[\pi \times 14.6 - \frac{6 \times \frac{1}{16}}{0.84}\right]} = 1.40 \text{ ins.};$$

$$b_1 = \frac{2.23 \times 144}{9.80\left[\pi \times 8.65 - \frac{6 \times \frac{1}{16}}{0.26}\right]} = 1.27 \text{ ins.}$$

7. *Method for Volute Pumps.* In the design the problem has been to find the dimensions of a turbine pump, one in which a diffuser has been used in which u is reduced to u_3 by the enlarging channel. As a result of this, u has been made quite large and c_2 has been small. This is due to the assumption of $\beta = 85°$, or some large value, and $\alpha_2 = -60°$. If now there is not sufficient space to reduce u to u_3, or if the diffuser is to be eliminated, then u is made small by selecting a large negative value for α_2 and a small value for β, as shown in Fig. 463. Such a pump is usually employed when the head is low. In this case the velocity u from the impeller should be radial and it is to be changed in the whirlpool chamber to u_3 by gradually enlarging the width and by the increase in the circumference. When the volute is reached the velocity is v_r in a tangential direction. The loss then is

$$L_v = \frac{(u_3 - v_r)^2}{2g}.$$

8. *Form of Impeller and of Vane Curves.* Using the results of the latter method the following data will apply to the pump:

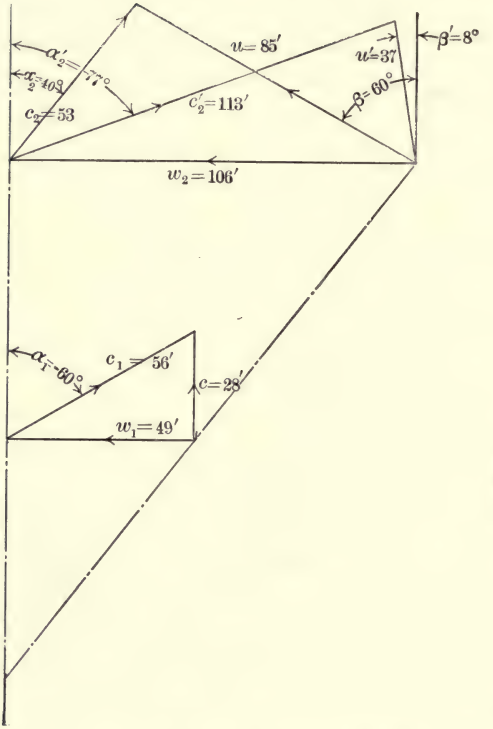

FIG. 463.—Diagram for Volute Pump and Low-head Turbine Pump.

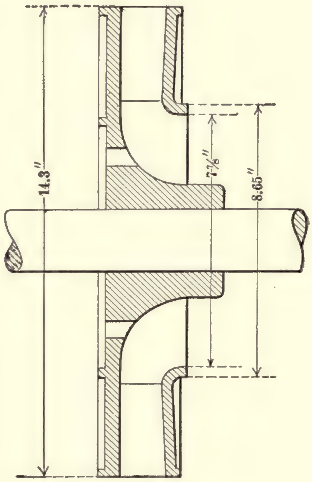

FIG. 464.—Section of Impeller.

$$w_2 = 63.8 \text{ ft. per sec.}$$
$$w_1 = 36.4 \text{ ft. per sec.}$$
$$c = c_r = 9.8 \text{ ft. per sec.}$$
$$c_1 = 37.8 \text{ ft. per sec.}$$
$$b_1 = 1.27 \text{ ins.}$$
$$c_2 = 6.2.$$
$$u = 60.3.$$
$$u_r = 5.26.$$
$$b_2 = 1.40.$$
$$D_1 = 8.65 \text{ ins.}$$
$$D_2 = 14.6 \text{ ins.}$$
$$\alpha_2 = -33°.$$
$$\alpha_1 = -75°.$$
$$\beta = 85°.$$

The data will be first applied in laying out an impeller which has pure radial action, such as used in Figs. 423, 429, 430. The cross-section will be laid out as shown in Fig. 464. The shaft diameter will be investigated after the weight of the impeller has been determined. The next important step is to lay out the vanes of the impeller and diffuser. There are several methods which may be employed.

In Fig. 465 the vane is drawn as a parabola. At A the

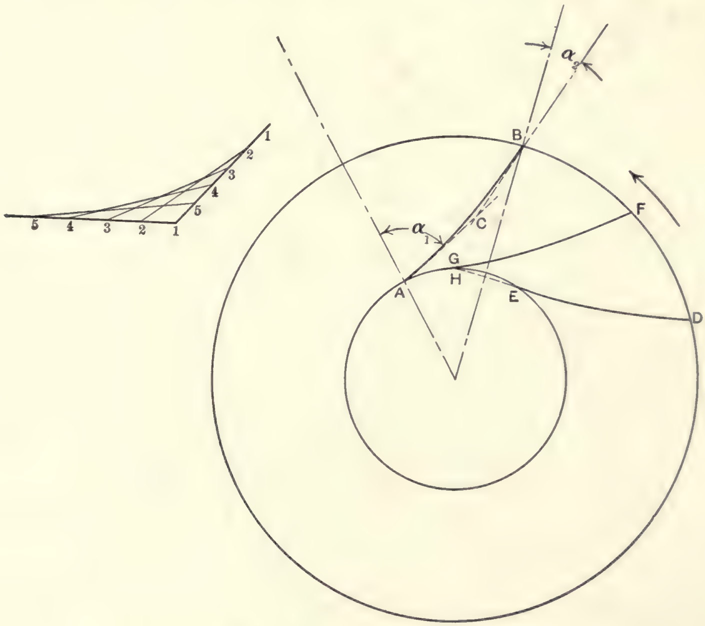

FIG. 465.—Parabolic Vanes.

angle α_1 is laid off from the radial direction and then the point B is so chosen that the line making the angle α_2 with the radius will intersect the line from A at C, about a mean radius between r_1 and r_2. These lines are tangents to the vane curve at entrance and exit and to put in the curve between the points, the lines AC and CB are divided into the same number of parts and lines are drawn connecting similar points, the top point of AC being connected to the top of CB, etc.

These lines are tangents to the parabola and hence their envelope will be the desired curve. When DE is drawn at one-sixth the circumference from BA it is seen that there is considerable change in angle from A to E, and hence an extra vane GF is put in. Since these vanes converge so rapidly as seen at H there is some danger of interference with the flow of water. Consequently these vanes are usually so drawn that

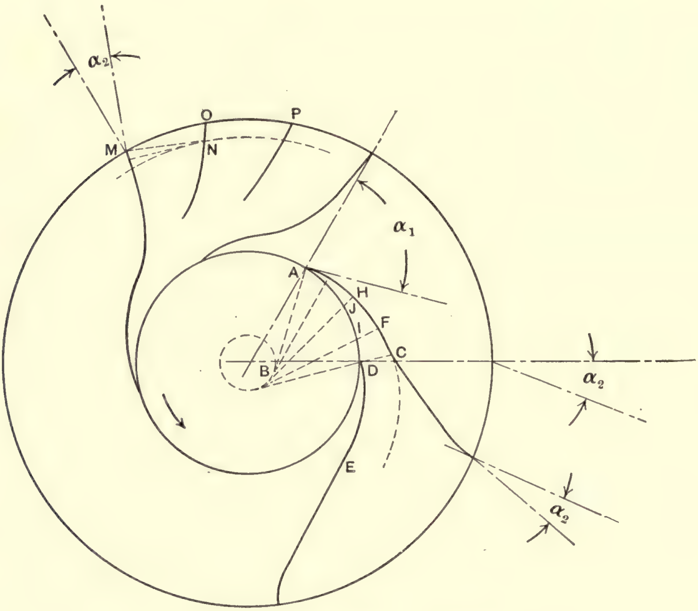

FIG. 466.—Involute Curves.

there is a portion of the vane just opposite to the corner of the next vane which is parallel to it.

Such a result may be obtained by using the involute as the curve at entrance and exit when possible. At A, Fig. 466, the angle α_1 is laid off as before and a perpendicular AB to this line is drawn. The circle tangent to this perpendicular will be the base circle of the involute AC. Another involute drawn from D will be parallel to the first involute, because the curves at the points C and D have the same centers of curvature.

If the involutes were carried further they would be parallel and at the constant distance CD apart. At the point C the water is moving parallel to that entering at D and consequently there is no tendency for the water at the entering corner to be affected by the interference of the previous vane. At F and H the water is so moving that it is parallel to water entering the impeller directly below it at I and J, when the water at

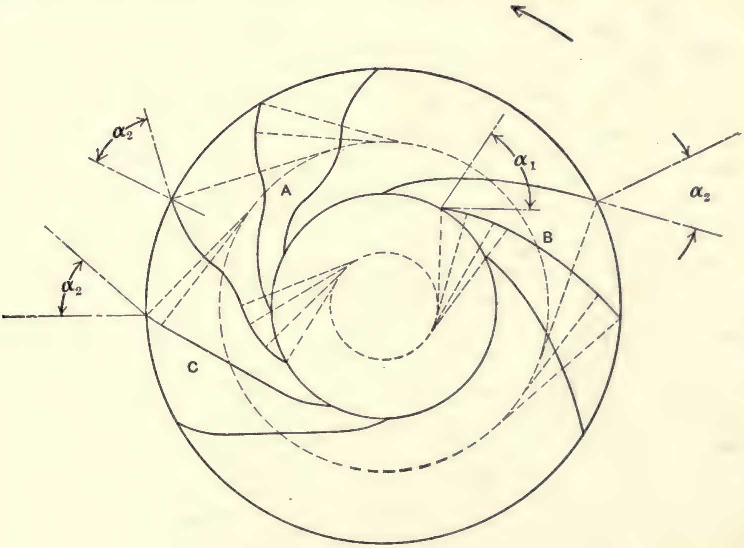

FIG. 467.—Involute Curves.

I and J is entering at an angle α_1 to the radius at the point in question.

When the same method is used at outflow, it is found that the circle tangent to the perpendicular MN, perpendicular to the line at an angle of α_2 with the radius, has its point of tangency N within the space left for one vane. This would mean that there would be nothing gained by the use of the involute. Intermediate vanes O and P are sometimes introduced to keep the outflow in the proper direction and by placing O so that it falls between N and M the action of outflow is similar to that at inflow. These vanes are only carried in a

portion of the distance toward inlet as it is not desired to obstruct the inflow. The involutes at inflow and outflow are connected by a curve as at M, or a tangent as at C.

When α_2 has a larger negative value the involute at outflow takes the form shown in Fig. 467. At A the two involutes are so far apart that a reverse curve has to be used in joining them, while at B a curve of the same form and curvature could be used. At C the involute at outlet is only used at the tip

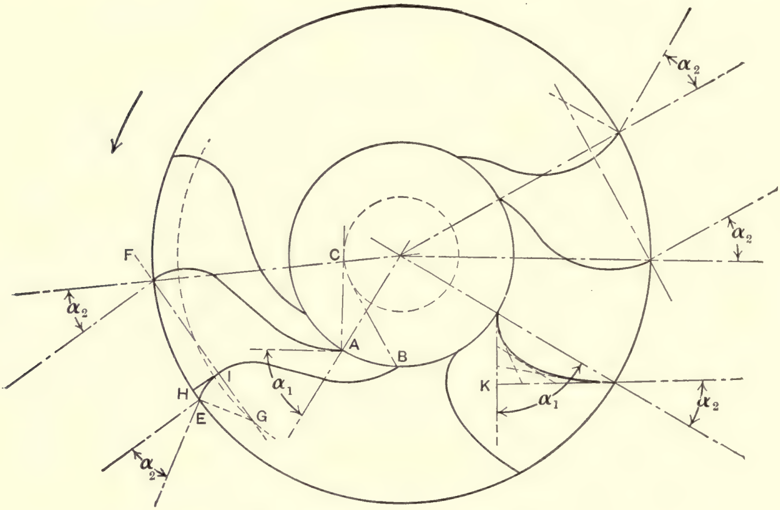

FIG. 468.—Circular Arc Vanes.

while a tangent is used between the parts. The form at B is quite common.

In Fig. 468 positive values of α_2 are shown as well as the method of using circular arcs. If perpendiculars to the lines of flow (at angle α_1 to radius) are erected at A and B these intersect at C. The point C may be used as a center for the arc from B to D. This curve at D is parallel to the curve A at the point A and hence there is no tendency to interfere with the flow. The perpendiculars at E and F at outflow intersect at G, which is the center for the arc at outflow. The centers at outflow and inflow lie on circles from the center

of the wheel and these are dotted in. After finding the centers C and G and the radii CA and GE, these other curves are quickly drawn in by using the dotted circles on which the centers must lie.

One set of vanes shows how the circular arcs are joined by tangents while another shows the use of a curve. At K a parabola is constructed showing the method of using that curve.

If the line HI is drawn in the direction of flow from I it appears as if there might be a dead space EHI where no water would flow, but this would be filled with eddies causing loss. If the change in form is too sudden this may happen, but it is to be remembered that this water is under pressure and it would consequently gradually enlarge if the curvature is not too great.

The curves drawn in the preceding figures are the center lines of the vanes and the half thickness is added to each side of the center line, the end being drawn to a sharp point.

Mixed Flow Pumps

The figures shown are applicable to pumps in which there is a pure radial action. If it is expedient to have some axial action at the center on account of a desire to keep the outer diameter small, the vane is carried into the center, as shown in Fig. 469. In this pump the peripheral speed at inlet changes for the various parts of the vane. At a the speed is w_{1a} at b, w_{1b}; at c, w_{1c}, etc. The velocity c should be the same at all points and in a direction normal to the various peripheral velocities. To have this result α_1 must vary over the inlet edge of the vane.

Before carrying out the steps used in design it will be well to consider the action at entrance. Suppose the number of vanes is assumed and the thickness of metal used for them. If in Fig. 469 the inlet edge be divided into four parts by the points 1, 2, 3, 4, 5, and the middle points a, b, c, and d be marked, the lines of flow aa'', bb'', cc'', dd'' may be approximated. These stream lines or lines of flow may be considered

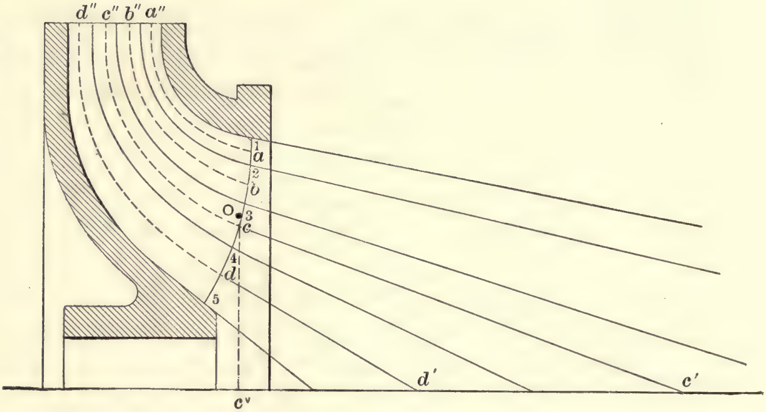

FIG. 469.—Section of Mixed Flow Impeller.

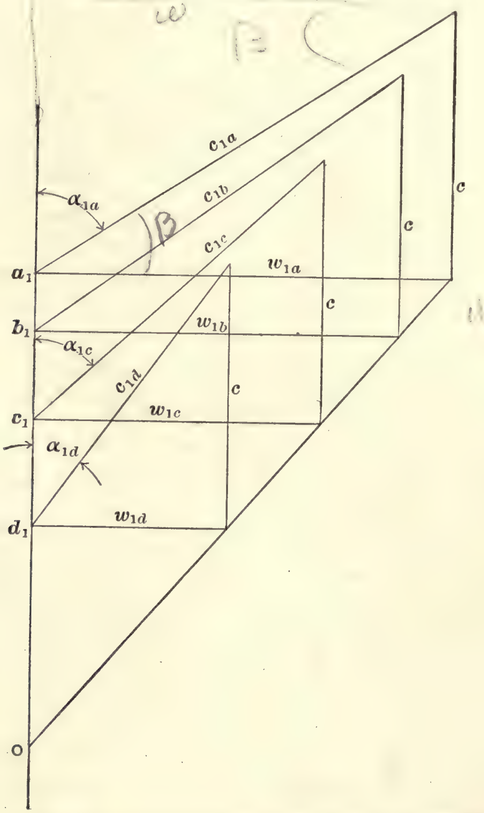

FIG. 470.—Velocity Diagram for Various Inflow Points.

to be on conical surfaces at the points of entrance, and if tangents are drawn from the points a, b, c, d to the axis of rotation, these lines are the elements of the various conical surfaces.

The peripheral speed at the points a, b, c, d may be found as shown in Fig. 470 where oa, ob, oc, etc., represent the radial distances to the various points. w_{1a}, w_{1b}, w_{1c}, etc., will represent the various peripheral velocities. These are known from w_2, since $w_{1x}=w_2\dfrac{r_{1x}}{r_2}$.

From Eqs. (3) and (4),

$$p_d-p_b=\frac{w_2^2}{2g}-\zeta_{ex}\frac{u^2}{2g}+\frac{c_1^2}{2g}-\zeta_v\frac{c_1^2}{2g}-\frac{c_2^2}{2g}-\frac{w_1^2}{2g}-\zeta_{ent}\frac{c^2}{2g}$$

$$=\frac{w_2^2}{2g}-\frac{c_2^2}{2g}-\frac{w_1^2}{2g}+\frac{c_1^2}{2g}-\zeta'_{ex}\frac{u^2}{2g}-\zeta_{ent}\frac{c^2}{2g}.$$

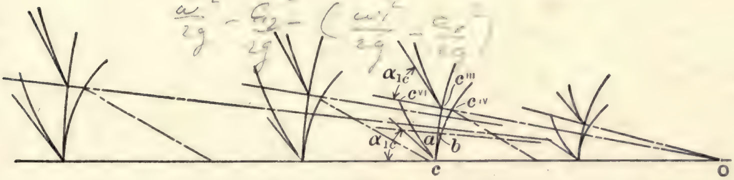

FIG. 471.—Angular Relations on Developed Conical Surfaces.

Now in the pump considered p_d-p_b is the pressure difference between that in the discharge space and that in the suction space. This is constant for all of the suction chamber and hence it is constant for all points of entrance. w_2 and c_2 are constants for all divisions of the pump and the losses may be considered as constant. Hence

$$c_1^2-w_{1x}^2=\text{constant}.$$

If c is to act at right angles to w_{1x} the expression above is equal to c. c is therefore a constant and the various values of c_1 and α_1 at a, b, c and d are found as shown in Fig. 470 by making the vertical distances all equal to c.

Fig. 471 is now constructed by developing the various conical surfaces and drawing on these developments the various vanes. The construction for point c is given as illus-

trating the method for all of the points. From the center o, Fig. 471, oc is laid off equal to cc' of Fig. 469, and with this as a center the arc cc''' is drawn. The arc cc^{iv} is now drawn of radius cc^{v} from Fig. 469 and c^{iv} is determined by the number of vanes used in the circumference (twelve assumed). If this curve is then rectified and placed on the arc of the conical surface, the pitch distance at the point c (Fig. 471) is found. The angles α_{1c} from Fig. 470 are now laid off from the radial lines at c and c''' and then the involute cc^{v1} is drawn in and from this the perpendicular distance between the vanes $c'''c^{v1}$ is found. This distance less the thickness of the vanes is the net depth of the passage or $c'''c^{v1} - t = d_c$.

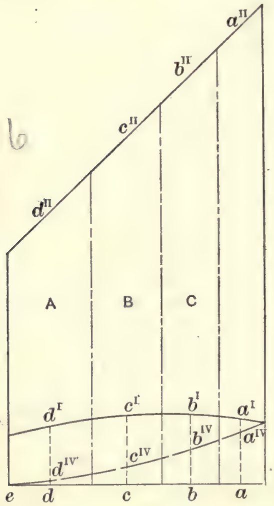

FIG. 472.—Curves of Area and Quantity.

This is done at each point and then the lengths of the outflow edge are rectified as in Fig. 472, giving the line a, b, c, d, as the length of the outflow edge. If now the depths found at the various points be laid off perpendicular to the above line at their respective points a figure $aa'd'd$ is found which represents practically the outflow area. This area is not quite equal to the outflow area as this is a warped surface which cannot be developed. Now the quantity of water flowing through any element of length is

$$dQ = c_1 d_c dl,$$

or

$$Q = \int (c_1 d_c) dl.$$

This means that Q is represented by an area in which the ordinates are $c_1 d_c$ and the abscissæ are l. Hence if the depths d are multiplied by the corresponding velocities c_1 from Fig. 470 and the product is laid off at the various points of the edge

in Fig. 472, the figure $add''a''$ is found, the area of which represents the quantity of water flowing.

Now in the formula (when inflow is at 0°),

$$\text{Work} = uw_2 \sin \beta.$$

The work per pound of water is the same at all parts of the impeller. Hence when this is put into the form

$$\text{Work} = Qwu^2 \frac{r_2}{r_1} \frac{A_{ex}}{A_1} \sin \beta \sin \alpha_1,$$

where r_1, A_1, and α_1 change for different parts of the impeller, the quantities r_1, A_1 and α_1 should refer to a point at the center of gravity of the outflow area. To show this suppose that the area of discharge is divided into a series of elements ΔA_{ex}, such that the quantity discharged is the same in each. Then $u\Delta A_{ex} = c_{1x}\Delta A_{1x} = K$ and

$$\text{Work} = \Sigma wc_{1x}\Delta A_{1x}\, u^2 \frac{r_2}{r_{1x}} \frac{\Delta A_{ex}}{\Delta A_{1x}} \sin \beta \sin \alpha_{1x}.$$

Now c_1 approximately varies as r, as may be seen from Figs. 470 and 472 and sin α_1 is practically constant.

$$\text{Work} = \Sigma K_{1x}\left(\frac{K''}{r_{1x}\Delta A_{1\alpha}}\right) = K\Sigma \frac{1}{r_1 \Delta A_{1\alpha}}.$$

Now $\Sigma r \Delta A_{1x}$ is the static moment of the inflow area and for that reason it equals $r_{c.g}A_1$, or the work is that required by the water if it all entered at the radius of the center of gravity of the actual area.

To find the center of gravity of $aa'd'd$ lay off the curve $d^{1v}c^{1v}b^{1v}a^{1v}$, found by multiplying each ordinate aa', bb', etc., by the distance from the left-hand end of the figure and using these as ordinates for the new curve,

$$aa^{1v} = aa'' \times ae,$$

$$bb^{1v} = bb' \times be,$$

$$\cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot$$

The area of this curve is the static moment of the original

area about the left corner. This area divided by the area of the original curve will give the distance of the center of gravity from the left corner,

$$\text{c.g.}=\frac{ed^{1v}c^{1v}b^{1v}a^{1v}}{dd'a'a}=\frac{\int ec\times d_c\times dl}{\int d_c\times dl}.$$

In designing a mixed flow impeller the method of procedure is as follows: Assume for a given design the radius at the center of gravity, as r_0, Fig. 469, and with u_2, w_2, c, and α_1, known for the point r_0, the constructions of Figs. 462 or 463 will give the distance d_0 and the velocity c_{1_0}. The length of outflow edge is now found approximately by the formula,

$$l=\frac{Q}{nc_{1_0}d_0}.$$

This is then used in Fig. 469 and a curve drawn equal to this in length. After the curve is assumed the various points are used as shown in Figs. 469, 470, 471, and 472, and the actual area and center of gravity is found. If now the center of gravity is slightly different from that assumed, or if the area of the quantity curve is greater than $\frac{Q}{n}$, the length or shape of the outflow edge is altered to give the correct values. Several trials will give the desired c.g. and quantity.

Since there is a slight change in the angle α_1 and moreover since the paths are of different projected lengths the surface of the vane is rather complex. The inlet edge is often contained in a radial plane so that the curve, Fig. 469, is seen in its true length. At other times the curve is in a plane which does not pass through the axis and sometimes the inlet edge is a non-planar curve. In the second case the projection of the curve on a plane perpendicular to the axis would give a straight line passing one side of the axis while in the last case the projection would be a curve.

Since the curves of the vanes at inlet have angles α_1 differing slightly from each other, and since the paths to the outlet

are of varying length the outlet line may not be parallel to the elements of the outlet cylinder, although by properly selecting the shape of the paths this may be accomplished.

Fig. 473 shows a first set of curves for a wheel. Fig. 469 and the cross-sectioned part of Fig. 473 are revolved projections of the vanes as if the complete vane was in a radial plane. The stream lines are not in the position shown. To obtain their correct position the following method is used: If the quantity curve of Fig. 472 be divided into equal parts, say four, by the lines A, B, and C, these lines determine the positions along the outflow edge at which divisions could be placed in the passage, so that each part would carry one-fourth of the quantity passing through the impeller. At the outlet edge the divisions will be spaced equally because c_2 and α_2 are the same at all points. If now the constructions of Figs. 470 and 471 be made for each of the points a, b, c, d, and e of Fig. 473, the true shape in projection of these lines may be found. The points at the half pitch across the bucket have been used to get better results. The points will all lie on a radius 15° from OR in the end view of the figure. Consider the part marked C of Fig. 471. The point of the vane on a radial plane 15° from the entering edge of the involute vane will be at a distance ab from involute to circle at the edge. This point is at a distance $ab=cc'$ from point c on the element of the cone, but when this cone is wrapped into position the true position of the point will be on an element 15° from the original point in the end view. Hence if c' is projected over to c'' in the end view and this is swung to the 15° radius, the point c''' is determined. This is carried over to the side view and c^{1v} is then determined by the intersection of this projecting line and a perpendicular to the axis from c'. In the same manner cc_1' is made equal to $c'''c^{v1}$ and c_1^{1v} is found from the point c_1''' in the end view, which is determined by swinging c_1'' over to the 30° radius. The intersection of the two projecting lines gives c_1^{1v}. The same operation is used for the other points. From the first two points of these various lines in the end view, the remainders of these lines are sketched

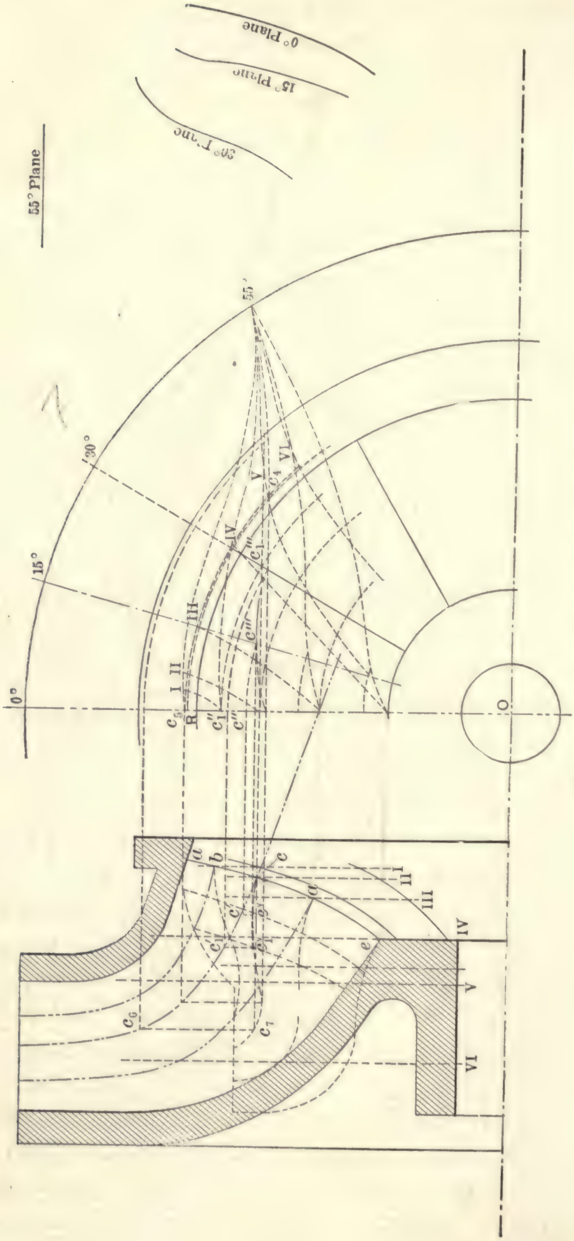

FIG. 473.—Construction of Vane Curves.

in for this view and these are all brought to the same element of the discharging cylinder and α_2 is made the same for each.

To construct the true shape of the various stream lines in the first view from the curves just sketched in, the following method is used: such a point as c_4 is revolved to c_5, carried over until it strikes the stream line from c at c_6, this is projected down until it cuts the projection line from c_4, giving the true point c_7. This is done for all stream lines, and if the shape of any is too complex or has too much reverse curvature, a new set of lines is assumed. After these two projections are made, sections of the vane form should be made at various angles for the purpose of seeing that the vane will have the proper shape in all directions. Such an operation, similar to fairing ship curves, is important. The curves seen at the right of Fig. 473 are the radial plane intersections. Those shown as heavy dotted lines in the end view are those on planes I, II, III, IV, V, and VI perpendicular to the axis. These latter curves are those used in constructing the core boxes from which the moulds for casting are made. In the above there has been no thickness of vanes shown. After finding the curves for one side of the vane a similar operation could be used for the other face.

Area Curve through Bucket

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After the vanes are determined and drawn in, it is well to find the area at various points along the middle line of the bucket (center of gravity of various areas) and if the lengths between various centers be laid off as a base line, Fig. 474, and the areas at these points are used as ordinates, the area of this figure, by the theorem of Pappus, is the volume of the vane bucket.

The curve so constructed should gradually increase or decrease. A curve with considerable change, as shown by the dotted line, is objectionable, as this means frequent changes in velocity $\left(v_x = \dfrac{Q}{A_x}\right)$ and changes in velocity are usually accom-

panied by loss. If such a curve as the dotted one is found, the axial width should be changed to bring it into the conditions shown by the solid line or the vanes are thickened up

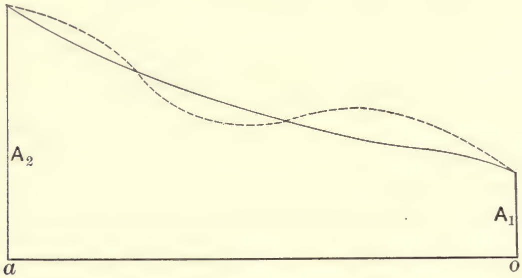


FIG. 474.—Curve of Passage Area.

by the use of back vanes, as shown in Fig. 475. These are necessary in the figure shown to keep the velocity more nearly constant, as the distance between the vanes is too great at the center.

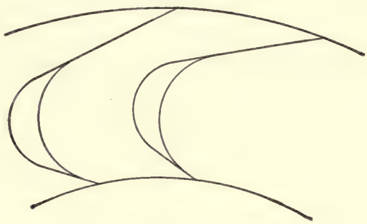


FIG. 475.—Back Vanes.

ABSOLUTE PATH OF WATER

The area of the curve, Fig. 474, between the inlet point and any other point is the volume of the bucket to that point and since A_1c_1 is the quantity of water entering the bucket per second, this volume divided by A_1c_1 is the time taken for a particle of water to move from the entering edge to the point considered,

$$t_x = \frac{\text{vol. }_x}{A_1c_1} = \frac{\text{area }_x}{A_1c_1}.$$

To find the absolute path of a particle of water passing over the vane *av* of the Fig. 476, the times taken to pass to the points v, x, y, z, and a are found, as shown, from a figure similar to Fig. 474. The points v, x, etc., are then carried in radially to v', x', y', etc., and from these points the distances $v'v_1'$,

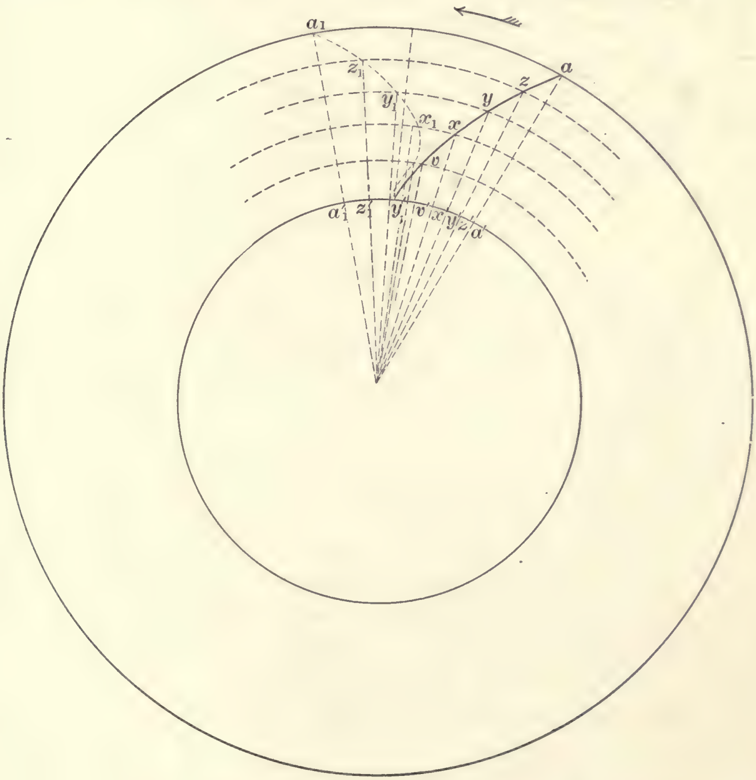


Fig. 476.—Absolute Path of Water.

$x_1'x'$, $y'y_1'$, . . . , equal respectively to w_1t_v, w_1t_x, w_1t_y, etc., are laid off. These points determine the positions of the radii when the water reaches the point in question. Hence, if from x an arc is drawn intersecting the radial line from x_1' this determines the point x_1 of the absolute path. In this manner a_1, z_1, y_1, x_1, and v_1 are found, giving the absolute path of the water through the impeller. This path should be a smooth

curve without sudden changes of curvature. The tangent to it represents the absolute direction of the water at any instant, hence at outflow the angle formed with the radius is β, while at entrance the angle is α.

9. *Diffusion Chamber.* Having the impeller designed the next step is to design the diffusion chamber. The angles of the vanes of the diffuser are β at entrance and α_d at discharge. Since the water is to discharge into the volute casing and travel in that chamber in a tangential direction it is advisable to make α_d as large as possible. The velocity of discharge v_d is given by the equation,

$$v_d=\frac{Q}{b_d\left[\pi D_d-\dfrac{ut_d'}{\cos\alpha}\right]}\cos\alpha_d.$$

When α_d is made large D_d must be large and t_d must be made very large if v_d is made small. This cannot in general be done, and in many cases α_d is made small, being 0° in some cases. This gives

$$v_d=\frac{Q}{b_d[\pi D_d-ut_d']}.$$

D_d may be so large in this case that if t_d is increased over the t at exit from the impeller, the net area, even after the enlargement of t' to give a better flow, to care for the supporting bolts or to make a passage as is done in the Worthington pump, is so large that v_d is quite small.

In Fig. 477 six vanes of the diffuser have been drawn as circular arcs with α_d and β as the angles at the two ends. On one of the center lines a plain vane has been drawn with its sharpened end, on a second one the vane has been constructed to make a channel for a supply to the next stage for balancing, while the third shows the enlargement of the vane to make a more gradual change in area and to form a place for the supporting bolt. The center line may be drawn if desired as an involute on the small base circle shown dotted.

Although the discharge in the last two cases is at right angles to the direction desired in the volute casing, the velocity

has been reduced to such a small value that even if this whole velocity head were lost the amount would be small. Moreover, the head lift for each impeller of the pumps using diffusers is so great that the percentage loss due to this radial outflow is very small. For this reason the volute casings in these pumps are not made as volutes but are concentric circular paths

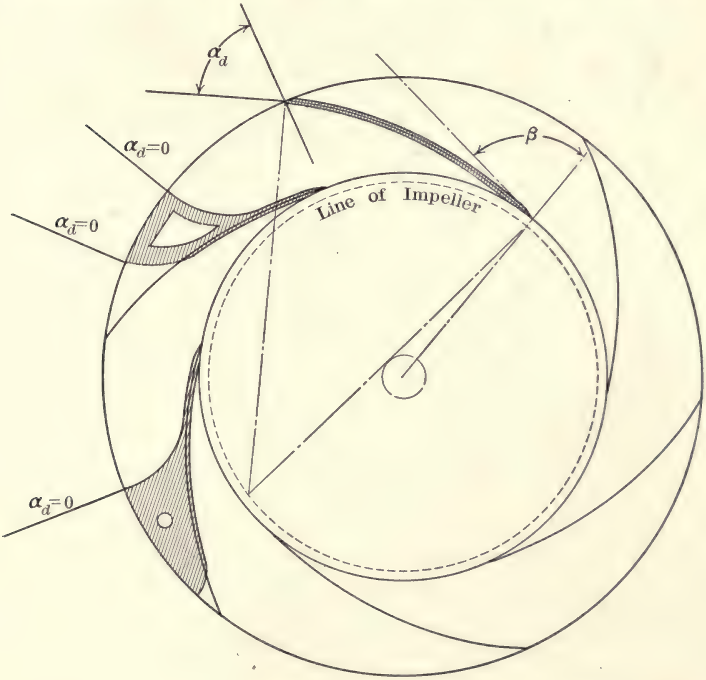


FIG. 477.—Diffuser.

uniting at the top of the pump as shown in Figs. 437, 438, and 441.

In the problem considered assume $D_d=15\frac{1}{2}+6\frac{1}{2}''=22''$, $b_d=b_2=1.40''$, $\alpha_d=0°$, $z'=4''$.

$$v_d=\frac{2.23\times 144}{1.40(\pi\times 22-6\times 4)}=5.3 \text{ ft. per sec.}$$

If b_d is increased to $1\frac{1}{2}b_2$ or $2.00''$ the velocity is changed to

$$v_d=3.5 \text{ ft. per sec.}$$

If this velocity is entirely lost in impact the loss is

$$L_d = \frac{v_d^2}{2g} = 0.31 \text{ ft.},$$

or less than $\frac{1}{4}$ of 1% of the head per stage.

10. *Volute Casing.* The volute casing of ordinary volute pumps is designed so that the velocity of the water is the

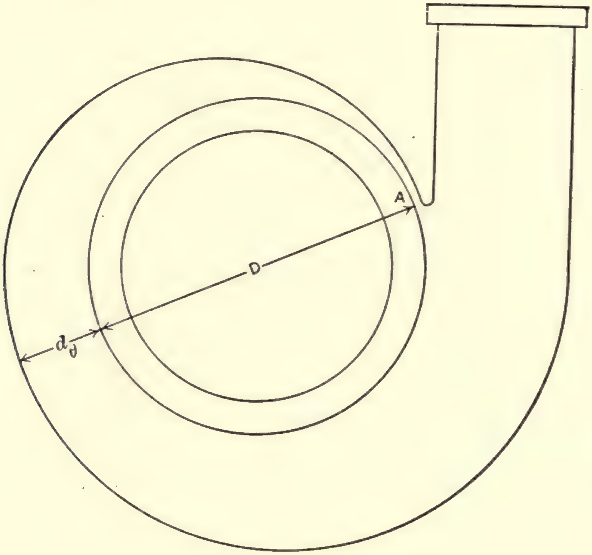


FIG. 478.—Volute Casing.

same at all portions. The quantity coming off from unit length of whirlpool chamber is

$$\text{Unit } Q = \frac{Q}{\pi D}.$$

The quantity passing at any point at the angle θ from A, Fig. 478, is

$$Q_\theta = \frac{Q}{D\pi} \frac{\theta D}{2} = \frac{Q\theta}{2\pi}.$$

If the velocity of this water is assumed as v_v, the area at the point is

$$A_\theta = \frac{Q\theta}{v_v 2\pi} = \frac{\pi d_\theta^2}{4} \text{ (if of circular section),}$$

$$d_\theta = \frac{1}{\pi}\sqrt{\frac{2Q\theta}{v_v}} = K\sqrt{\theta}. \qquad \left[K = \frac{1}{\pi}\sqrt{\frac{2Q}{v_v}}\right].$$

If A is rectangular and of constant width w in direction of the axis, the dimension in the direction of the radius is

$$d_\theta = \frac{Q\theta}{v_v 2\pi w} = K\theta. \qquad \left[K = \frac{Q}{2\pi v_v w}\right].$$

Neumann points out that the first assumption of a circular section gives the limiting curve of Fig. 478 the form of a parabolic spiral, while in the second case the limiting curve is an involute.

11. *Shaft Design.* The weight of the impellers should be computed from the drawing made, as in Fig. 464, and from previous experience the length of the shaft to care for these impellers is known. A diagram, such as shown in Fig. 479, is made, giving weights and positions between supports.

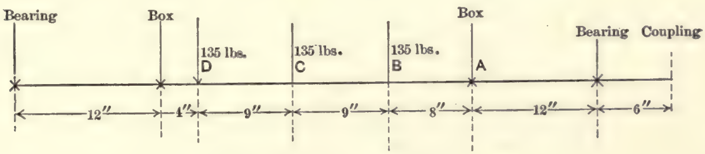


FIG. 479.—Load Diagram.

Considering the beam as a simple beam, the bending moment at the point A is 0 and the twisting moment T is

$$T = \frac{100 \times 33{,}000 \times 12}{1000 \times 2\pi} = 6300 \text{ in.-lbs.,}$$

from B to C,

$$T = \tfrac{2}{3} \times 6300 = 4200.$$

The reactions due to the loads of 135 pounds at each impeller will be $229\frac{1}{2}$ pounds at the left and $175\frac{1}{2}$ pounds at

the right. The shear diagram will pass through zero at the second load, hence the moment will be a maximum at this point.

$$M = 229\tfrac{1}{2} \times 13 - 135 \times 9 = 1768 \text{ in.-lbs.}$$

Under the last impeller the moment is

$$M = 175\tfrac{1}{2} \times 8 = 1404 \text{ in.-lbs.}$$

The combined moment is

$$T_c = \sqrt{M^2 + T^2}.$$

At B this is

$$T_c = \sqrt{1404^2 + (6300)^2} = 6400;$$

at C,

$$T_c = \sqrt{1768^2 + (4200)^2} = 4550.$$

For the shaft diameter,

$$6400 = \tfrac{1}{2} S_t \frac{\pi d^3}{16}$$

$$= \frac{5000 \pi d^3}{16},$$

$$d = 1.87.$$

As will be seen, the critical speed for such a combination of discs on a shaft demands a larger diameter, and for that reason a diameter of 2 has been assumed before the investigation for critical speed.

The thrust bearing will be investigated at this point. The thrust is caused by the impact and pressure on the inlet area of the pump and the unbalanced pressure on the shrouding or sides of the impellers. Using the dimensions from Fig. 464,

Diameter at hub	$= 3\tfrac{1}{2}$ ins.
" inner clearance ring	$= 8\tfrac{1}{2}$ ins.
" " " back	$= 7$ ins.
" discharge	$= 14.6$ ins.
" outer edge of entrance	$= 7\tfrac{7}{8}$ ins.

$$p_a-p_b=\frac{w_2^2}{2g}-\frac{c_2^2}{2g}-\frac{w_1^2}{2g}+\frac{c_1^2}{2g}-\zeta'_{ex}\frac{u^2}{2g}-\zeta_{ent}\frac{c^2}{2g}.$$

$w_2=63.8$ ft. per sec.

$w_1=36.4$ ft. per sec.

$c_2=\ 6.2$ ft. per sec.

$c_1=37.8$ ft. per sec.

$u=60.4$ ft. per sec.

$c=9.80$ ft. per sec.

$$p_a-p_b=\frac{1}{2g}[63.8^2-6.4^2-36.4^2+37.7^2-0.15\times 37.7^2]=62.5 \text{ ft. head.}$$

If the water leaks on each side to the center, as seen in Fig. 419, this pressure difference acts on each side and consequently there is no unbalanced pressure from this source. If there was leakage only on one side, say, the back of the impeller, then it would be assumed that the pressure in space would be a portion of the 62.5 feet, as the water leaks out at center into the suction. Suppose this be assumed to be two-thirds of 62.5, or 40 feet approximately.

The pressure to the right is then

$$P=\frac{40\times 62.5}{144}\left[\pi\frac{14.6^2}{4}-\pi\frac{7^2}{4}\right]=2180 \text{ lbs.}$$

The force from the impact of the water to the left is

$$\frac{Wv}{g}=\frac{wAc^2}{g}=\frac{62.5\frac{\pi}{4}(7\tfrac{7}{8}^2-3\tfrac{1}{2}^2)9.8^2}{144\times 32.2}=50 \text{ lbs.}$$

If there are openings at the center of the impeller to relieve the pressure and if leakage occurs on each side, the impact is all that has to be cared for.

The total pressure to be carried on the thrust bearing in this pump is

$$\Sigma P=3\times 50=150 \text{ lbs.}$$

This would require the area,

$$A=\frac{P}{p_1}=\frac{150}{50}=3 \text{ sq.ins.}$$

If there are four collars on the 2-inch shaft the approximate height of the collars to give the proper area will be

$$h=\frac{A}{n\pi D}=\frac{3}{4\times\pi\times2}=\tfrac{1}{8}\text{ in.}$$

The collars will be made $\frac{5}{16}$ inch high for easier machine work.

Critical Speed

If a shaft which is deflected slightly is caused to revolve, it is subject to centrifugal force due to the weights turning at the angular speed ω. When the speed is low the shaft has a chance to bend to the elastic curve due to the weights of the body and the various parts will rotate about their figure centers, or geometrical centers, which are about the same. When, however, the speed is increased this bending cannot occur so rapidly and the shaft, pulleys and weights may be assumed to rotate around the axis of the bearings. Following Reynolds' method as given by Stanley Dunkerley in his excellent paper published in the Philosophical Transactions for 1894, Part A, assume the shaft and disc as shown in Fig. 480. The load per element

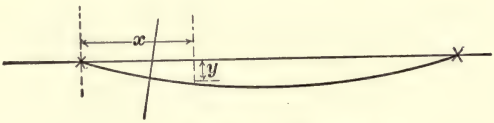


FIG. 480.—Whirling Shaft and Pulley.

of length of the shaft when turning with the angular velocity ω at which the weight effect, but not the mass effect, can be neglected is

$$\text{Load}=\frac{Awdx}{g}\omega^2y=Ldx,$$

where A = area of shaft in sq.ft.
w = weight of 1 cu.ft.
g = acceleration of gravity.
y = deflection.

Now $\text{Shear}=\int Ldx=V,$

where L = load per foot and V = shear.

Moment $=\int V\,dx=M$, but $M=EI\dfrac{d^2y}{dx^2}$ where E equals modulus of elasticity in pounds per sq.ft. and I is the moment of inertia of the cross-section in feet^4. Hence $V=EI\dfrac{d^3y}{dx^3}$, $L=EI\dfrac{d^4y}{dx^4}$. The equation of the elastic curve then, when the speed is such that the deflection is caused by the centrifugal force and not by the weight, is

$$\frac{Aw}{g}\omega^2 y=EI\frac{d^4y}{dx^4}.$$

This occurs when the effect of weight is eliminated or the shaft is moving at such a speed that whirling occurs.

The equation may be written as

$$m^4 y=\frac{d^4y}{dx^4},$$

where $m=\sqrt[4]{\dfrac{Aw\omega^2}{gEI}}$.

This integrates into

$$y=A'e^{mx}+B'e^{-mx}+C'e^{mix}+D'e^{-mix},$$

or $$y=A\cosh mx+B\sinh mx+C\cos mx+D\sin mx.$$

A, B, C, D are the constants of integration.

To eliminate these constants there are several known conditions. At bearings, $x=0$ or l, and $y=0$; for fixed bearings $\dfrac{dy}{dx}=0$, at a change of loading x, y, and $\dfrac{dy}{dx}$ for the equations on one side of the load are equal to the respective equations which hold on the other side of this point. The equation above has been derived from the condition that $L=\dfrac{Aw}{g}\omega^2 y$ for the curve and this only holds between concentrated loads. At each concentrated load or bearing for a shaft with several bearings there is a new equation. At the concentrated loads

$$V_L-V_R=-\frac{W}{g}\omega^2 y=EI\frac{d^3y}{dx^3}-EI\frac{d^3y'}{dx_1{}^3}.$$

When there is a disc on the wheel which is deflected by the bending of the shaft into the position shown in Fig. 481 the centrifugal action of the disc tends to right the disc and straighten the shaft.

For an element dm of the disc at distance r from the center, of which the components are v, t and b, the component of the centrifugal force tending to right the disc is $\frac{dm}{g}v\omega^2$ and its arm is t. The moment of this is $tv\omega^2\frac{dm}{g}$ or $\frac{dy}{dx}v^2\omega^2\frac{dm}{g}$,

since
$$\frac{t}{v}=\frac{dy}{dx}.$$

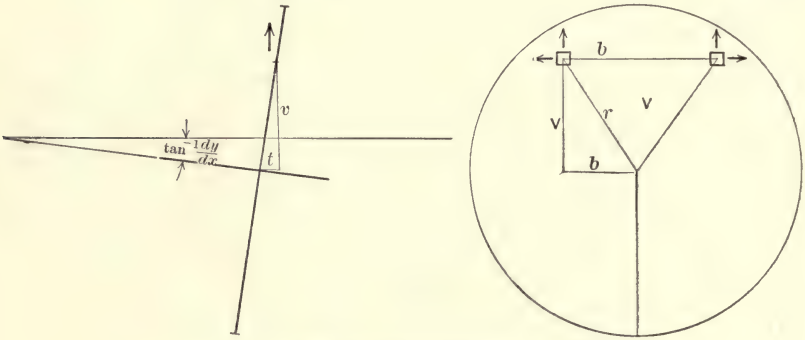


FIG. 481.—Rotating Disc.

The total righting moment is

$$M=\Sigma\frac{\omega^2}{g}\frac{dy}{dx}v^2dm$$

$$=\frac{\omega^2}{g}\frac{dy}{dx}\int v^2dm.$$

Now
$$v^2=r^2-t^2-b^2,$$

hence
$$M=\frac{\omega^2}{g}\frac{dy}{dx}(I_p'-I_t'-I').$$

I_t' is very small,
$$I_p=2I',$$

hence M, the moment due to centrifugal force tending to right the disc, is

$$M=\frac{\omega^2}{g}\frac{dy}{dx}I'.$$

In passing a disc the difference between the moments on the two sides is

$$M_L-M_R=-\frac{\omega^2}{g}I'\frac{dy}{dx}=EI\left[\left(\frac{d^2y}{dx^2}\right)-\left(\frac{d^2y'}{dx_1{}^2}\right)\right].$$

This is another condition in passing a load which eliminates another constant. These conditions will furnish sufficient equations to eliminate the constants.

Dunkerley in his extensive article takes up the different cases arising in practice and determines the critical speed. The student is referred to the article for the methods of solving the equations, but the simple case of a plain shaft is given here to show how these are worked out.

For a simple shaft, the equation for y is

$$y=A\cosh mx+B\sinh mx+C\cos mx+D\sin mx.$$

Now when

$$x=0 \text{ or } l,\ y=0,$$

and when

$$x=0 \text{ or } l,\ M=0, \text{ i.e., } \frac{d^2y}{dx^2}=0.$$

$$\frac{dy}{dx}=mA\sinh mx+mB\cosh mx-mC\sin mx+mD\cos mx.$$

$$\frac{d^2y}{dx^2}=m^2A\cosh mx+m^2B\sinh mx-m^2C\cos mx-m^2D\sin mx.$$

$$O=A+C.$$

$$O=A\cosh ml+B\sinh ml+C\cos ml+D\sin ml.$$

$$O=A-C.$$

$$O=A\cosh ml+B\sinh ml-C\cos ml-D\sin ml.$$

$\therefore\ A=0,\ C=0.$

$$B\sinh ml+D\sin ml=0.$$

$$B\sinh ml-D\sin ml=0.$$

Subtracting

$$D\sin ml=0.$$

$$D=0 \text{ or } ml=\pi.$$

Adding

$$B \sinh ml = 0.$$
$$B = 0 \text{ or } ml = 0.$$

The condition which is possible is

$$ml = \pi,$$

$$\frac{Aw}{g}\frac{\omega^2}{EI}l^4 = \pi^4,$$

$$\omega = \frac{\pi^2}{l^2}\sqrt{\frac{gEI}{Aw}} = \frac{N\pi}{30},$$

$$N = \frac{30\pi}{l^2}\sqrt{\frac{gEI}{Aw}}.$$

In the case of a disc on a shaft as shown in Fig. 481 the following results if the mass of the shaft is not considered, although the resistance to deflection is taken into account:

$$EI\frac{d^4y}{dx^4} = 0,$$

$$EIy = \frac{A}{6}x^3 + \frac{Bx^2}{2} + Cx + D.$$

$$x = 0, \qquad y = 0.$$

$$x = 0, \qquad M = \frac{d^2y}{dx^2}EI = 0.$$

$$x = c, \qquad y = y'.$$

$$x = c, \qquad \frac{dy}{dx} = \frac{dy'}{dx'}.$$

$$x = c, \qquad \frac{d^3y}{dx^3} - \frac{d^3y'}{dx'^3} = -\frac{W}{gEI}\omega^2 y.$$

$$x = c, \qquad \frac{d^2y}{dx^2} - \frac{d^2y'}{dx'^2} = -\frac{I'}{EI}\frac{\omega^2}{g}\frac{dy}{dx}.$$

$$x = l, \qquad y' = 0.$$

$$x = l, \qquad EI\frac{d^2y'}{dx^2} = M \quad = 0.$$

These give the following equations in order

$$D=0,$$
$$B=0,$$
$$\frac{Ac^3}{6}+Cc=\frac{A'c^3}{6}+\frac{B'c^2}{2}+C'c+D'.$$
$$\frac{Ac^2}{2}+C=\frac{A'c^2}{2}+B'c+C'.$$
$$A-A'=-\frac{W}{gEI}\omega^2\left(\frac{Ac^3}{g}+Cc\right)=a\left(\frac{Ac^3}{6}+Cc\right),\text{ if } a=\frac{W}{gEI}\omega^2.$$
$$(A-A')C-B'=\frac{I'}{EI}\frac{\omega^2}{g}\left(\frac{Ac^2}{2}+C\right)=K^2a\left(\frac{Ac^2}{2}+C\right),\text{ if } K^2=\frac{I'}{W}.$$

K^2 is the square of the radius of gyration $\frac{I'}{W}$.

$$O=\frac{A'l^3}{6}+\frac{B'l^2}{2}+C'l+D'.$$
$$O=A'l+B'.$$

Eliminating the constants $A, B, C, D, A', B', C', D'$, from these equations Dunkerley obtains the following equation, calling $l-c=c'$:

$$a^2K^2+3\left\{\frac{al}{cc'}-aK^2\left(\frac{1}{c^3}+\frac{1}{c'^3}\right)\right\}-\frac{9l^2}{c^3c'^3}=0.$$

$$K^2=\frac{\frac{3l}{acc'}\left[\frac{3l}{ac^2c'^2}-1\right]}{1-\frac{3}{a}\left(\frac{1}{c^3}+\frac{1}{c'^3}\right)}=\frac{\frac{3l}{acc'}\left[1-\frac{ac^2c'^2}{3l}\right]}{\frac{ac^2c'^2}{3l}-\frac{1}{l}\left(\frac{c'^3}{c}+\frac{c^3}{c'}\right)},$$

but $\qquad l=c+c'.$

$$K^2=\frac{\frac{3l}{acc'}\left[1-\frac{ac^2c'^2}{3l}\right]}{\frac{ac^2c'^2}{3l}-\left(\frac{c'}{c}+\frac{c}{c'}-1\right)}.$$

Now K^2 must be a positive quantity, since it equals $\frac{I'}{W}$.

Hence

$$1<\frac{ac^2c'^2}{3l}<\left(\frac{c'}{c}+\frac{c}{c'}-1\right).$$

Now $a=\frac{W}{gEI}\omega^2$ and these inequalities give the limiting speeds ω for the centrifugal force to act to produce whirling. These are only the limiting values beyond which there is certainly no whirling or beyond which it is known to be without the range of the critical speed. To get the critical speed the equation for K^2 is solved for a and then a is expressed in terms of ω^2, giving

$$\omega^2=\frac{3gEI}{2Wc^3}\left[\left\{1+\left(\frac{b}{1-b}\right)^3-\left(\frac{a^2}{1-b}\right)\right\}\right.$$
$$\left.+\sqrt{\left\{1+\left(\frac{b}{1-b}\right)^3-\frac{a^2}{1-b}\right\}^2+\frac{4a^2b}{(1-b)^3}}\right].$$

In this $a=\frac{c}{K}$=ratio of the distance from pulley to nearer bearing to the rectangular radius of gyration.

$b=\frac{c}{l}$=ratio of distance from pulley to bearing to the span (less than $\frac{1}{2}$).

Calling $\frac{3}{2}$ of the bracket above θ^2, Dunkerley gives

$$\omega=\theta\sqrt{\frac{gEI}{Wc^3}}.$$

In this equation I is the moment of inertia of the shaft section, W is the weight of the disc or pulley, c is the distance from the nearer bearing to the disc, θ is a factor which depends on the ratio $\frac{c}{l}$ and on the ratio $\frac{c}{K}$ where K is the rectangular radius of gyration.

Dunkerley then computes a table for θ for various values of a and b which has been reproduced in the form of a series of curves shown in Fig. 482.

Dunkerley considers the case of a shaft with three supports and also the case of a span with an overhanging end of length c. If the ratio of the short span to the longer span is

$\alpha_1\left(\frac{l_s}{l}=\alpha\right)$ and if the same symbol α be used for the ratio $\frac{c}{l}$, for the second case, he gives the specific speed for each in the form

$$\sqrt[4]{\frac{Aw\omega^2}{gEI}}l=K.$$

In these two cases the values of K depend on α and Dunkerley has computed the values of K for different values of α. These have been plotted in Fig. 482 so that ω may be found and from it N.

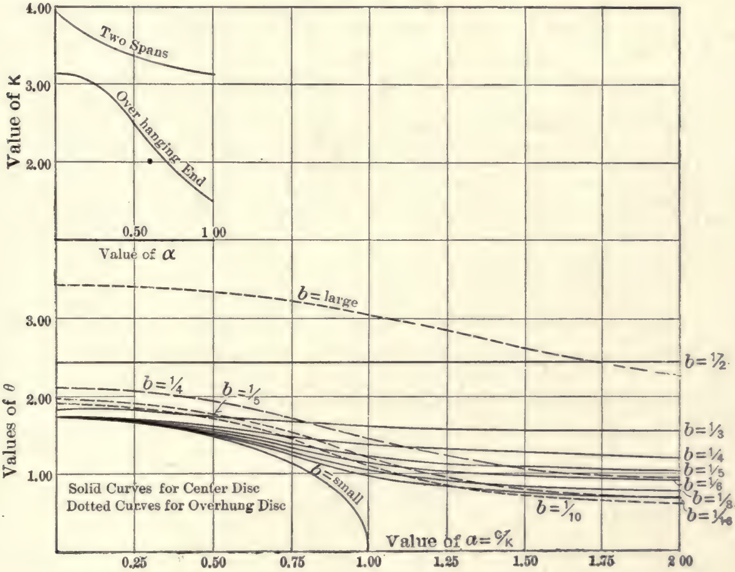


FIG. 482.—Dunkerley's Values for Shafts.

For the case of a disc on the projecting end of length c, the formula

$$\omega=\theta\sqrt{\frac{gEI}{Wc^3}}$$

holds for the disc independent of the shaft.

The value of θ, as in the case of the disc between supports depends on the value of $a=\frac{c}{K}$ and $b=\frac{c}{l}$. Dunkerley's values

of θ for four values of $\frac{c}{l}=b$ have been plotted for different values of $a=\frac{c}{K}$. The value of θ may be taken from the curves from which ω and N may be determined.

In the case of the double-span shaft it is seen that the value of K does not vary much from π, showing that the main effect of using several bearings is to shorten the span only. This shortening has considerable effect, although the continuous beam action is not important. If there must be an overhanging end of a shaft of fixed total length it may be shown that when $\frac{c}{l}=\frac{1}{3}$ or the bearing is $\frac{1}{4}$ the total length from the overhanging end the critical speed will be the greatest.

With the curves of Fig. 482 the critical speed can be figured for each disc, wheel or pulley on a shaft independently of each other and of the shaft. When these are computed the resultant speed is such that

$$\frac{1}{N_r^2}=\frac{1}{N_1^2}+\frac{1}{N_2^2}+\frac{1}{N_3^2}+\ldots,$$

where N_r is the resultant speed and N_1, N_2, N_3, etc., are the critical speeds due to any one part independent of the other parts. Hence

$$N_r=\frac{N_1N_2N_3N_4\ldots}{\sqrt{N_1^2N_2^2\ldots N_{n-1}^2+N_2^2N_3^2\ldots N_n^2+N_1^2N^2{}_3\ldots N_n^2+\ldots}},$$

or for the case in hand of three impellers and a shaft, suppose N_1 is the critical speed for the shaft, and N_2, N_3, and N_4 those for the impellers. The critical speed of the combination will be

$$N=\frac{N_1N_2N_3N_4}{\sqrt{N_1^2N_2^2N_3^2+N_2^2N_3^2N_4^2+N_3^2N_4^2N_1^2+N_4^2N_1^2N_2^2}}.$$

If these happened to be identical, the following would result:

$$N=\frac{N_1}{2}.$$

In the design of the shaft the stuffing boxes are considered to be the same as bearings on account of the closeness of fit. This gives for investigation of critical speed a shaft of three spans with an overhung disc. The shaft will be considered alone as a shaft with a 12-inch span and a 30-inch span, as no constants are given for shaft of three spans such as is shown. This gives the lowest speed of any combination for the shaft. All dimensions are in feet.

$$I=\frac{\pi d^4}{64}=\frac{\pi(2)^4}{64}=.7854\text{ in.}^4=\frac{0.7854}{12^4}\text{ ft.}^4.$$

$$E=29{,}000{,}000\times 144.$$

$$\frac{l_2}{l_1}=\frac{12}{30}=0.4.$$

$$K=3.4.$$

$$N_s=\frac{30K^2}{\pi l^2}\sqrt{\frac{gEI}{Aw}}=\frac{30\times(3.4)^2}{3.14\times\left(\frac{30}{12}\right)^2}\sqrt{\frac{32\times 29{,}000{,}000\times 144\times\frac{0.785}{(12)^4}}{\frac{\pi}{4\times 144}2^2\times 460}}$$
$$=12{,}500.$$

For the impeller discs the weights are taken as 90 pounds each and the radius of gyration $=\frac{D}{4}$. The separate discs give the following:

$$K=\frac{14.6}{4}=3.65.$$

Disc 1, $b_1=\frac{4}{30}=0.13,\quad a_1=\frac{4}{3.65}=1.1,\quad \theta_1=1.05.$

Disc 2, $b_2=\frac{13}{30}=0.43,\quad a_2=\frac{13}{3.65}=3.56,\quad \theta_2=2.1.$

Disc 3, $b_3=\frac{8}{30}=0.26,\quad a_3=\frac{8}{3.65}=2.2,\quad \theta_3=0.9.$

$$N_1=\frac{30}{\pi}\theta\sqrt{\frac{gEI}{Wc^3}}=\frac{30\times 1.05}{\pi}\sqrt{\frac{32\times 29{,}000{,}000\times 0.785}{90\times\left(\frac{4}{12}\right)^3\times 144}}=12{,}400.$$

$$N_2 = N_1 \frac{2.1}{1.05}\sqrt{\frac{c_1^3}{c_2^3}} = 12{,}400 \times \frac{2.1}{1.05}\sqrt{\left(\frac{4}{12}\right)^3} = 4580.$$

$$N_3 = 12{,}400 \frac{0.9}{1.05}\sqrt{\left(\frac{4}{8}\right)^3} = 3760.$$

For the disc on the end $K = \frac{6}{4} = 1.5$, $b = \frac{6}{12} = \frac{1}{2}$, $a = \frac{6}{1.5} = 4$, $\theta = 2.0$, $c = 6$, $W = 15$ lbs.

$$Nc = 12{,}400 \frac{2.0}{1.05}\sqrt{\left(\frac{4}{6}\right)^3\left(\frac{90}{15}\right)} = 31{,}300.$$

The resultant of these various speeds is given by the formula

$$\frac{1}{N_r^2} = \frac{1}{N_s^2} + \frac{1}{N_1^2} + \frac{1}{N_2^2} + \frac{1}{N_3^2} + \frac{1}{N_c^2}$$

$$= 64 \times 10^{-10} + 65 \times 10^{-10} + 210 \times 10^{-10} + 705 \times 10^{-10} + 10 \times 10^{-10}$$

$$= 1095 \times 10^{-10}.$$

$$N_r = 3000.$$

The speed N_3 is the factor which has the greatest control of the resultant. These terms depend on $\sqrt{I}$ for their values, so that if the diameter of the shaft was reduced to $1\frac{1}{2}$, the approximate value of N would be

$$N = 3000\sqrt{\left(\frac{1\frac{1}{2}}{2.0}\right)^4} = 1680,$$

a value so close to the actual speed that there might be considerable vibration and shock. This makes clear the reason for the increase in the diameter.

Centrifugal Pumps for Special Purposes

The centrifugal pump has been used for many years for the clearing of dry docks. Fig. 483 illustrates the equipment for one of the docks at the League Island Navy Yard. The units are 45-inch volute pumps built by Worthington; each is driven by a 450-H.P. motor and will handle an average quantity of 50,000 gallons per minute against heads varying

from 1 foot to 33 feet. As was pointed out with the test curves, the quantity decreases as the head increases, the speed remaining constant. The power usually reaches a maximum at an intermediate head, so that there is no danger of overloading the motor in any case. It is this feature which makes the centrifugal pump of value for this service.

The quantity of 50,000 gallons per minute, which is the usual way of rating centrifugal pumps, would mean 72,000,000 gallons per twenty-four hours, the method used in rating water-works pumps.

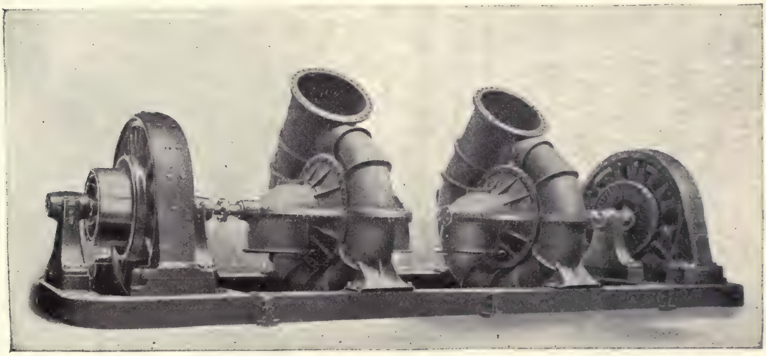


FIG. 483.—Dry Dock Units. (Worthington.)

These pumps for many years have been used extensively for this work and for draining the low lands of Holland and the marshes of Italy. Fig. 484 shows one of the stations built by Gwynne in 1876. There are eight pumps placed in pairs driven by compound engines. The boiler houses are located at the ends of the pump room. This arrangement shows a good plan and one well thought out. The pumps were of double-flow volute type handling 57,000 gallons per minute under a head of $7\frac{1}{4}$ feet. The impellers were 60 inches in diameter and two of them were driven by a $27\frac{3}{4}$-inch and $46\frac{5}{8}$-inch by 27-inch engine. The plant as shown in *Engineering*, was built for the Ferrara Marshes in northern Italy.

The pumps of that day gave efficiencies of 50 to 70 per cent

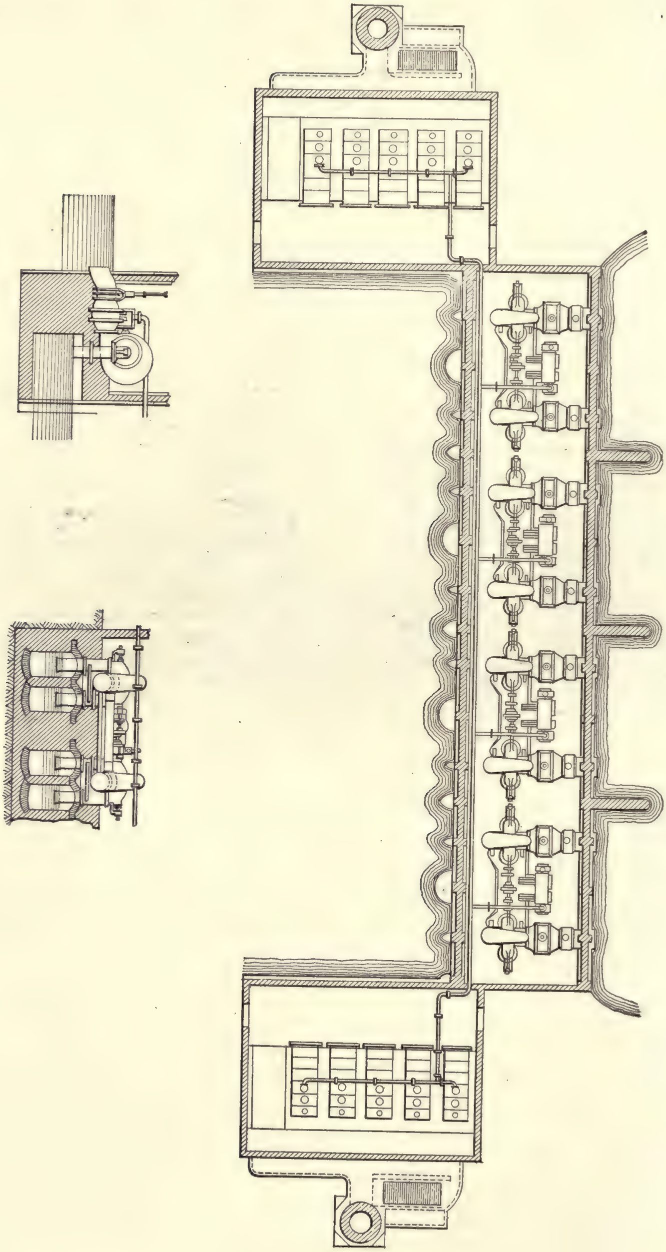


FIG. 484.—Station for Ferrara.

and were very reliable. The higher efficiencies obtained to-day are due to the better method of design.

One of the largest centrifugal pumps built in 1884 by Simpson & Co. for the London docks of the East and West India Companies, handled 46,400 gallons per minute.

The pumps shown in Fig. 485 are remarkable in that they handle 35,000 gallons per minute under the high head of 160 feet, requiring a 2000-H.P. motor to drive each of them. These pumps were of the turbine type on account of the high head

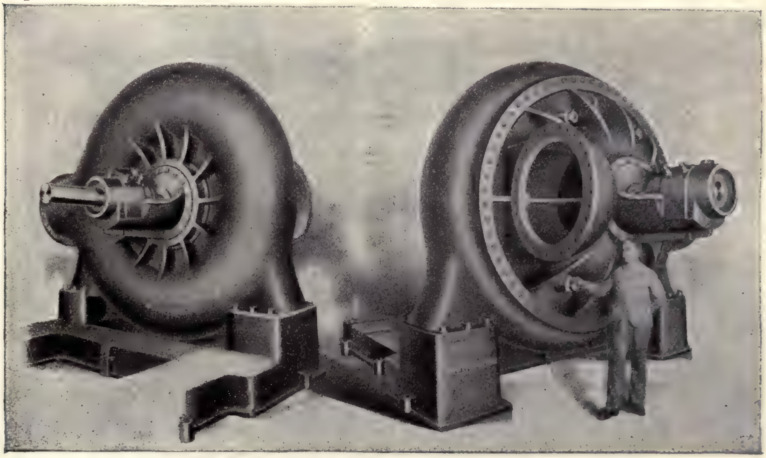


FIG. 485.—36-Inch Turbine Pump of Worthington.

and were used at the St. Louis exposition to supply water to the Grand Cascade. The volute casing of concentric form with an outlet at right angles is common with this type of pump. The figures show how part of the diffuser is cast with the volute casing on one side while the head containing the suction flange and elbow forms the other part of the diffuser.

Fig. 486 illustrates a casing of an R. D. Wood pump of 50,000 gallons per minute for condenser work. This is a 45-inch pump. Many pumps of this type are used with vertical axes. The figure illustrates the method used in forming the casings of pumps of large diameter. Much ingenuity is displayed at times in the methods of separating the parts.

The pump shown in Fig. 487 is one recently built by the Alberger Company for the Standard Oil Company. It is a turbine pump and consists of three units connected in parallel, as was the case in the tri-rotor volute pump. This pump is to handle 20,000,000 gallons per twenty-four hours. The form of the casing with the discharge outlets at right angles to the main direction of flow is seen here as in the other turbine pumps.

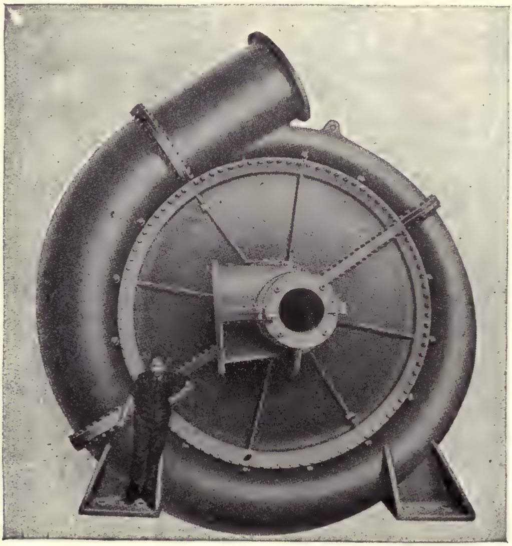


FIG. 486.—R. D. Wood 45-Inch Centrifugal Pump.

Another important use for the centrifugal pump is that of dredging channels. In this case, silt, sand and even rocks are raised with the water by the centrifugal pump and delivered into scows or settling basins on land through long flexible pipes carried on pontoons. The solid material settles out from the water and the water is returned to the stream. This method forms a very cheap and effective manner of dredging where possible. The pump has to be so constructed that the blades of the impeller may be easily replaced when broken or worn out. It is quite evident that the solid matter will

cause rapid wear of these parts. It is also important to have all passages direct and of ample size to pass large pieces of solid matter which may be carried through the pump.

Mr. F. B. Maltby, in the Transactions of the American Society of Civil Engineers, Vol. 54, gives an excellent description of dredges used on the Mississippi River and the machinery on them. Mr. Geo. Fowler gives a drawing of the type used in New York harbor in Vol. 31, p. 468, of the Transactions of

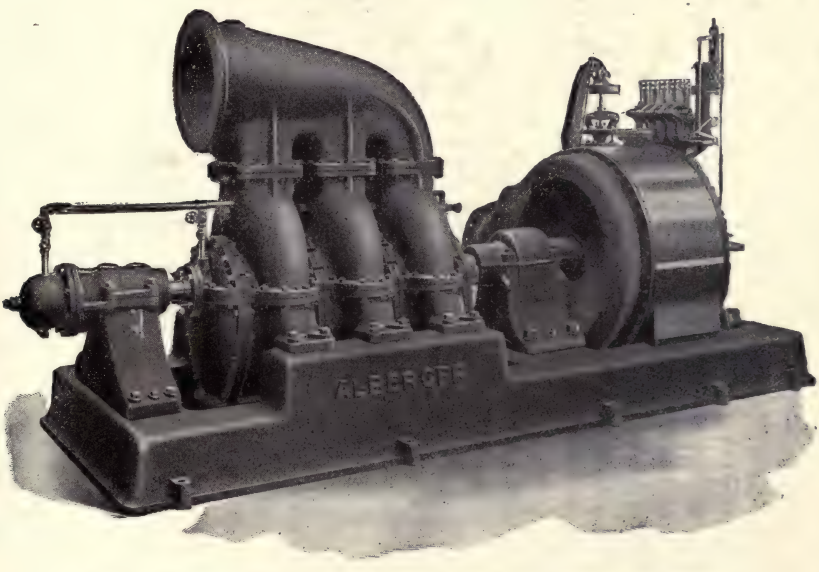


FIG. 487.—20,000,000 Gallon Alberger Multi-impeller Turbine Pump.

the American Society of Mechanical Engineers. From his description Fig. 488 has been prepared. The figure shows the method of attaching the vanes, tips and the large passages used with these pumps. Mr. Fowler reports that in dredging the New York Ship Channel a piece of shaft weighing 70 pounds was lifted and passed by a centrifugal pump, and at Yonkers, N. Y., an 8-inch pump on a wrecking boat lifted and passed a 35-pound piece of pig iron $11\frac{1}{2} \times 4\frac{3}{4} \times 3\frac{1}{4}$ inches. These dredge pumps are driven at a speed of about 1100 feet per minute at 178 R.P.M. Those used in New York harbor on

one contract were driven by 192-H.P. engines, delivering 10,000 gallons per minute. Mr. Fowler shows the detail of the end of the suction line. This end piece sinks in the soft silt or sand and the solid matter is lifted with the water. Extra openings permit water entering the suction pipe when the silt covers the mouth too much for proper action.

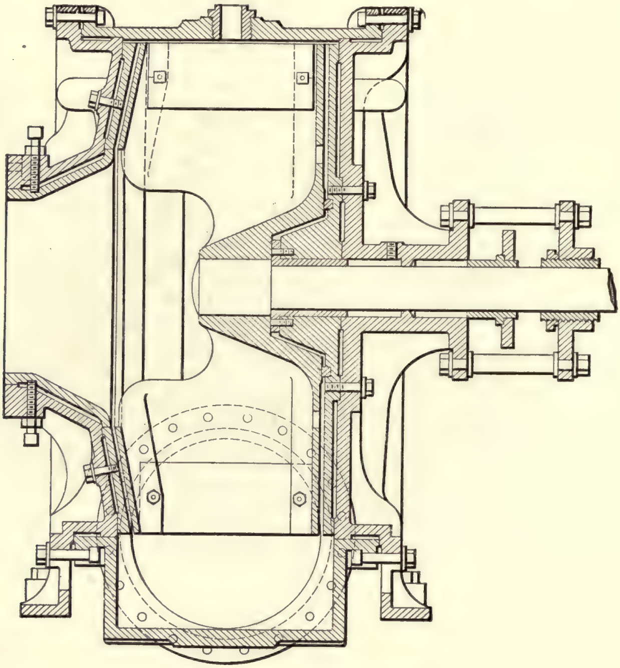


FIG. 488.—Dredging Pump.

One of the latest uses for centrifugal and piston pumps is that for the high-pressure fire service. The great fire hazards in the congested districts of trade in our large cities has made it necessary to build a separate high-pressure water supply system throughout these districts. The pumping stations may be placed on a river front, where an unlimited supply of water may be had, or in the center of the city, in which case a special supply line of large size is brought from one of the city reservoirs. The use of salt water in cities such as New York or

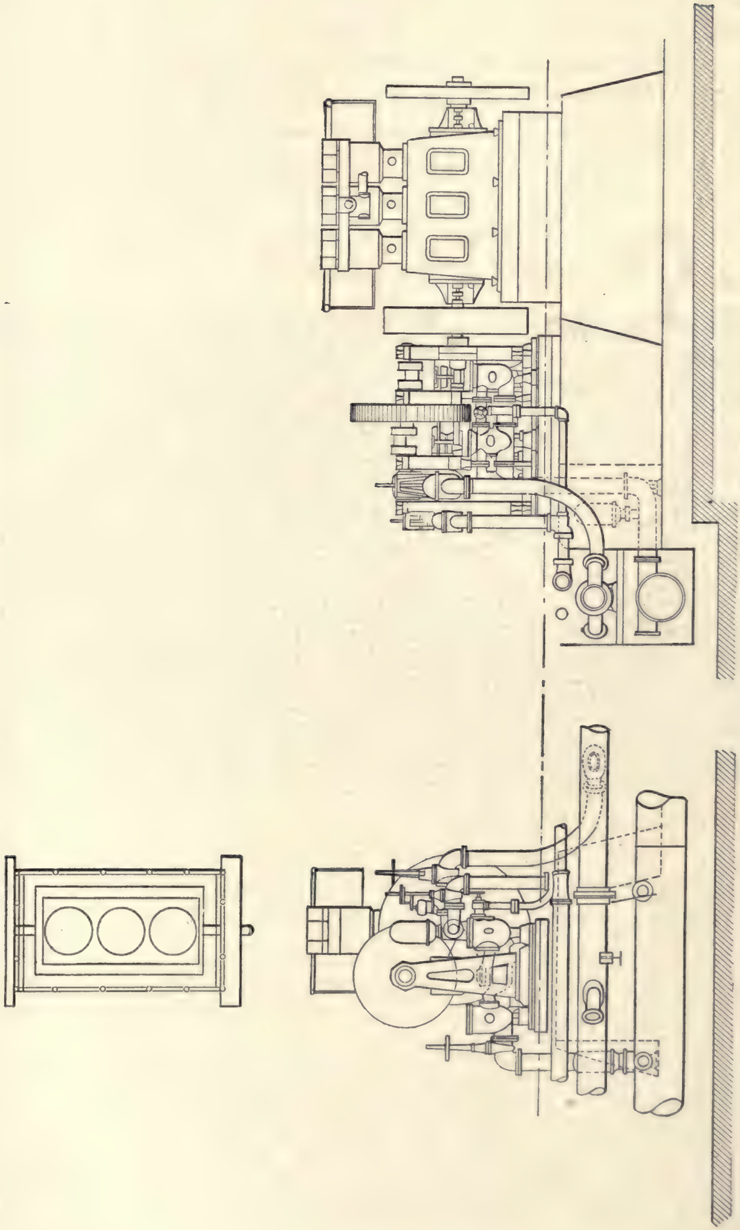


FIG. 489.—Pumps at Philadelphia.

Brooklyn is objected to on account of the damage done by salt water when it touches merchandise. For this reason in New York the supply is taken from the fresh-water mains. The use of water for fire service in such a city is a very small percentage of the total water used, so that this is not a very expensive method, and in the usual system employing fire engines the water is drawn from fire hydrants on the fresh water supply, so that the cost of water is not increased when city water is used with the new stations.

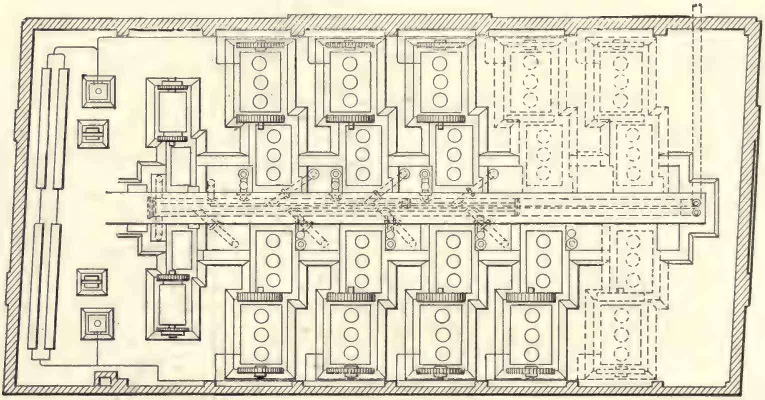


FIG. 490.—Philadelphia High-pressure Station.

From the pumping stations extra heavy flanged piping is carried in a network over the district. Special hydrants are used and in many cases fire lines are led into the various buildings. Valves must be used at frequent intervals to control lines leading into buildings or sections of the system, so that the waste of water could be prevented in case a wall should fall covering the valve controlling the branch leading to a burning building.

When a fire occurs the alarm is sent to the pumping station and the pumps are put into commission. One of the earliest of these stations was erected for the city of Philadelphia. The pumps were of the triplex form driven by gas engines. These are shown in Fig. 489. The gas engines are furnished with gas from a special city main. The engines

are started by compressed air held in storage tanks at one end of the building. There are several methods of igniting the charge, so that should one method fail another may be employed. The Westinghouse engines used in this station have always responded to the demands of the service. The Dean Triplex Pumps were built to give 1200 gallons per minute under 300 pounds pressure. The gas engines gave 280 H.P. each.

Fig. 490 shows the seven 1200-gallon double-acting pumps and the two 350-gallon pumps originally put in with the two air

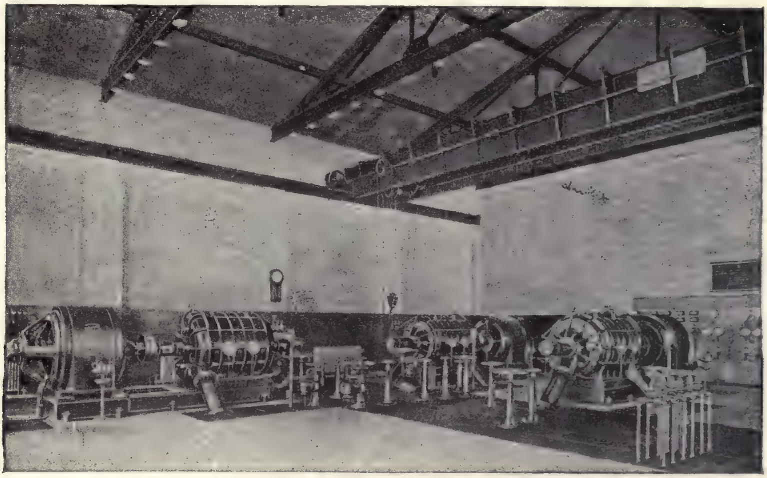
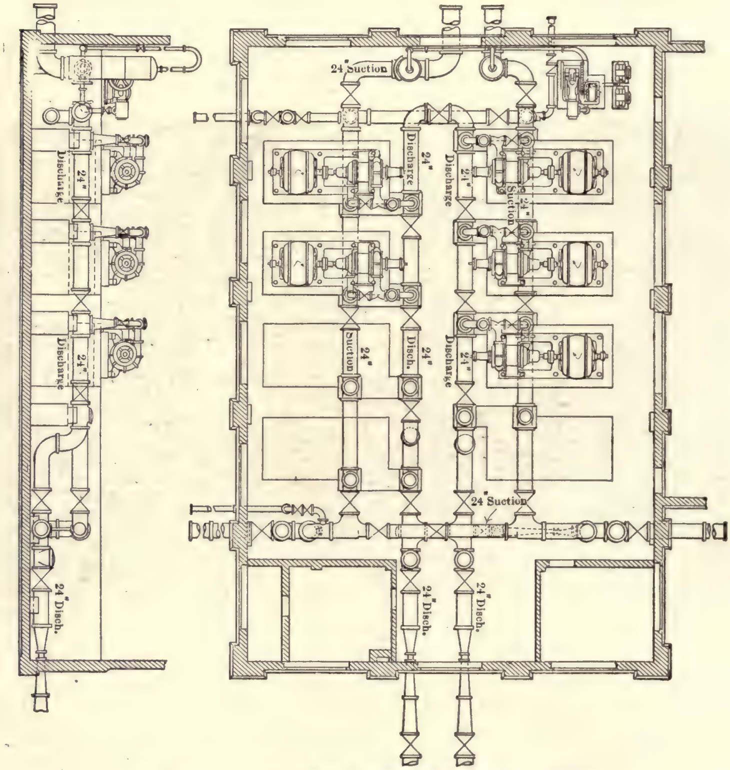


FIG. 491.—High-pressure Pumping Station, Brooklyn, N. Y.

compressors placed at the west end of the building. These air compressors are used to charge the air tanks, which are made up of heavy piping. The suction pipe enters from the river, passing between the two rows of pumps. It is connected to the various pumps by motor-operated valves. The pumps are started under no load and after the engine is operating properly the by-pass valve on the pump is closed, allowing the discharge to enter the pressure main. The pressure main is connected by a check valve with the city mains, so that there is always a pressure of 60 pounds in this line. Provisions are made for connecting fireboats to the pressure line in case

there should be a need for it. The gas engine is off centered from the pump to accommodate the gears.

In 1906 the same kind of system was introduced in Brooklyn, using turbine pumps driven by means of electric motors. The Brooklyn station is shown in Fig. 491. The five-



PLAN OF THE STATIONS, SHOWING PUMPS AND PIPING

FIG. 492.—New York High-pressure Station.

stage Worthington turbine pumps deliver 3000 gallons per minute under a head of 300 pounds. The test curve from these pumps is shown in Fig. 458. The use of electric motors for such stations must be safeguarded by the use of several cross-connected power houses, so that an accident to one will not endanger the reliability of the station.

One of the latest installations is that for New York city. There are two stations, one on East River at Oliver Street and one on the North River at Gansevoort Street. Each station has five five-stage Allis-Chalmers centrifugal pumps driven by Allis-Chalmers induction motors. The plan of the station is shown in Fig. 492, while the interior of one of the stations is shown in Fig. 493. This photograph shows the appearance of the pump

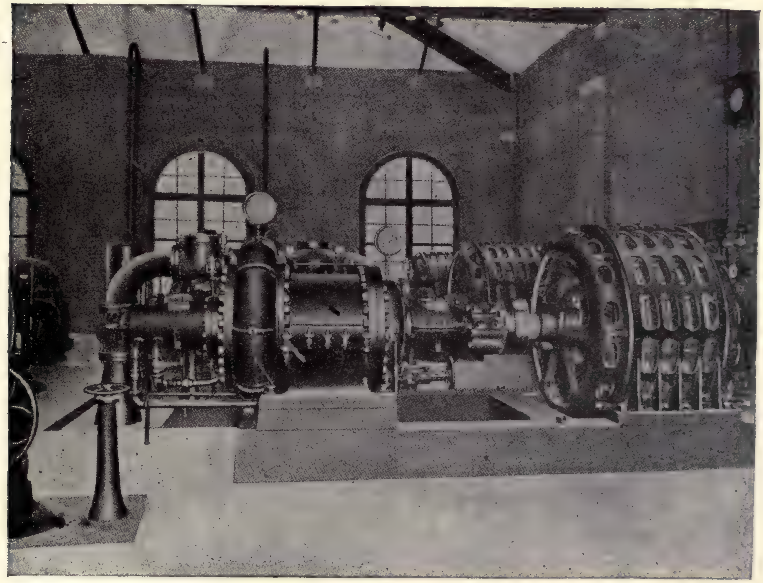


FIG. 493.—New York High-pressure Station.

with its outflow casing at the left hand end and the valve on the suction pipe at the right. The 3-phase, 25-cycle, 6300-volt, 740-R.P.M. induction motors are also seen. The pressure gauges give some idea of the proper operation of the pumps. The extended thrust bearing with its cooling pipes is seen at the left. A section of this pump was given in Fig. 454.

There are two suction pipes leading to the river at the top of Fig. 492. And in addition there are two fresh-water suction pipes entering from the city mains on the sides. The two air chambers on the river suction pipes are for the

purpose of steadying the flow. These are kept charged by the suction air pump shown at the upper right-hand corner. The suction pipes form a loop around the station. The discharge mains are also arranged in a loop and both lines are equipped with valves, so that any section of the pipe may be cut out without affecting the operation of the station. Venturi meters are used on the discharge to measure the water used by the system.

The discharge from each pump is controlled by a valve, so that as soon as the pressure in the main becomes greater than the predetermined amount, due to the decreased use of water, a special valve by-passes a portion of the discharge into the suction. In all of the central high-pressure stations the pressure is regulated according to the wishes of the fire chief, as he is in telephonic communication with the station at all times. The pressure is changed by changing the number of units in operation.

CHAPTER XV

MINE PUMPS

THE method of mine pumping has been changed very materially of late years. The early method introduced by Newcomen is the one which has been usually followed until recent times. Fig. 494 shows the method of using a long rod extending from the engine house to the pump barrels. When necessary to take off a pump in a side gallery a bell-crank lever was mounted at the side of the rod, one end of which was attached to the rod; the other, to the branch rod or pump.

When necessary to balance these long rods, beams were mounted above ground or in the rod shaft and on these counterbalance weights were attached.

By arranging the bell crank lever so that its pump operates on the stroke of the main pump on which no pumping is done, balancing may be accomplished.

These reduced the load to be moved by the piston, but the inertia of the system was greatly increased. Another method used at times was to place two pumps in the same shaft, balancing one set of rods by the other. Such a pump, built in Aix-la-Chapelle (Fig. 495), illustrates this method very clearly and gives some idea of the complicated system used. The figures illustrate engines with fly wheels although many are used without fly wheels, direct-acting connections being employed. Some of these pumps used in America are remarkable for their size and weight. The Mexican Union Pump of 1880, built with Leavitt jacketed cylinders, had 64×96 inches for H.P. cylinders and 100×102 for L.P. cylinder. A 36-foot fly wheel was used. The pumps were arranged at various points of the rods, there being fourteen different plungers. The rods were 2618 feet long, and there was so much con-

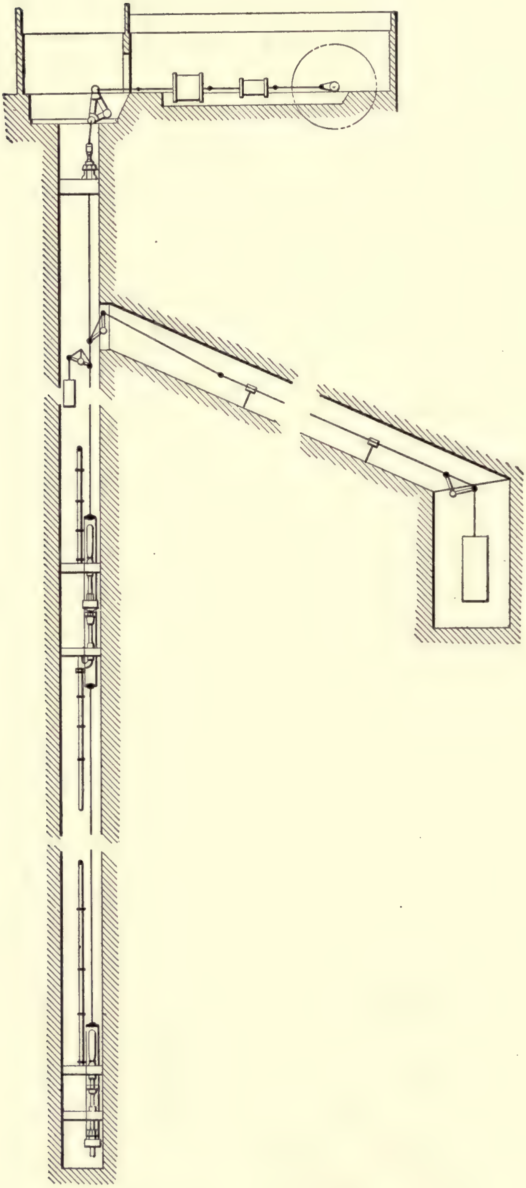


FIG. 494.—Mine Pump.

traction in the rods during action that the 10-foot stroke at the engine was reduced to 97 inches at the pumps. This system of

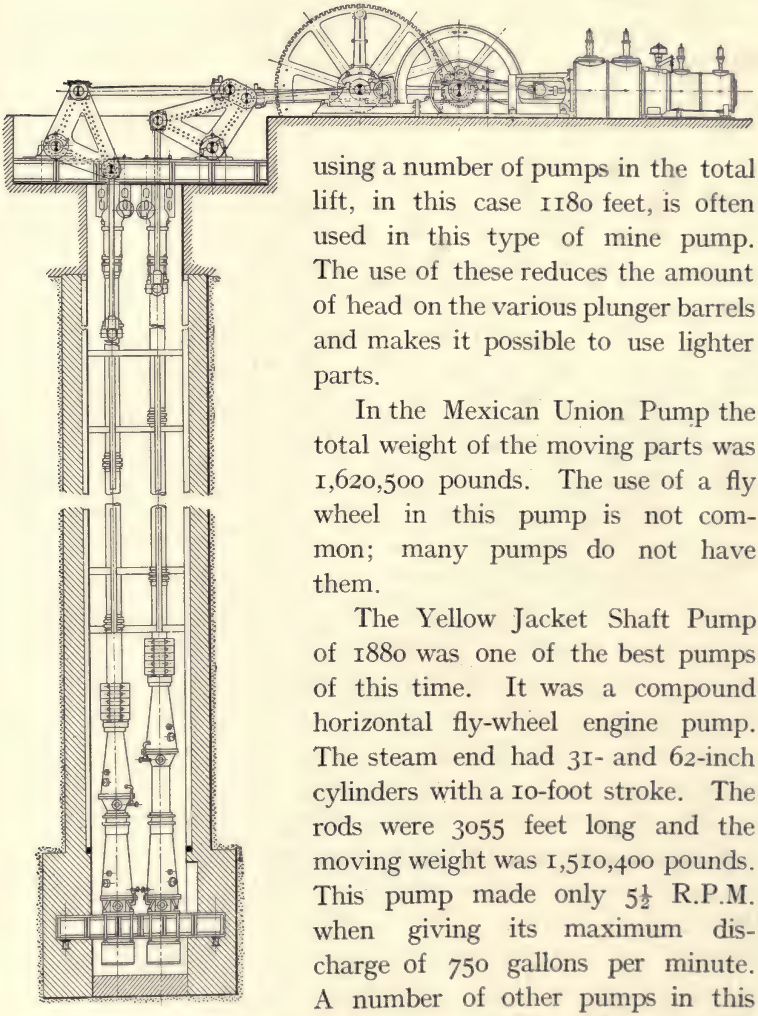


FIG. 495.—German Mine Pump.

using a number of pumps in the total lift, in this case 1180 feet, is often used in this type of mine pump. The use of these reduces the amount of head on the various plunger barrels and makes it possible to use lighter parts.

In the Mexican Union Pump the total weight of the moving parts was 1,620,500 pounds. The use of a fly wheel in this pump is not common; many pumps do not have them.

The Yellow Jacket Shaft Pump of 1880 was one of the best pumps of this time. It was a compound horizontal fly-wheel engine pump. The steam end had 31- and 62-inch cylinders with a 10-foot stroke. The rods were 3055 feet long and the moving weight was 1,510,400 pounds. This pump made only $5\frac{1}{2}$ R.P.M. when giving its maximum discharge of 750 gallons per minute. A number of other pumps in this district were built with compound cylinders 32×129 and 65×96 inches with no fly wheels and a total weight of moving parts of 1,437,900 pounds.

At the Consolidated California Shaft, a Davey Differential

Pump with steam cylinders 24×90 and 40×96, and pumps of 7 feet stroke lifted 500 gallons per minute. The moving parts with 2150 feet of pump rod weighed 860,000 pounds.

A number of such engines to handle 5400 gallons per minute against a head of 1152 feet cost $1,300,000 without foundations or installation. The cost of operation when 5040 gallons were lifted 1074 feet was $58,210 per month, or $677,440 per year.

In contrast to this cumbersome method, the method of installing independent pumps, Fig. 496, is to be mentioned. In this case, the pump is installed at a point where needed, and by the use of proper air chambers, the effect of the inertia in the water can be eliminated.

The pump may be made sufficiently strong and by using the express type of pump, or the centrifugal form, a sufficient capacity may be installed in a small space. The operation of such a pump must be by steam, compressed air, or electricity. The first method is out of the question for great depths, as the condensation of steam means much trouble; moreover, the radiation losses from the steam pipe would be excessive even if the use of high superheat brought the steam to the pump in a dry condition. The use of compressed air may be possible if a high grade compressor is used and the exhaust may be used for ventilation. The loss, however, in this case is one which must be considered carefully. The last method of the use of the electric motor is one which has many points of advantage.

Power may be brought to the motor in a very simple manner and the location of the apparatus is not difficult. The transmission loss with moderately high voltages and proper frequency is not great.

Fig. 496 shows a shaft in which water is raised from one pump at a great depth and discharged into another at a considerable height above it. The water is then lifted by a second pump through the remaining height. The express pump shown in the figure with its motor or a centrifugal pump uses a very small amount of space when compared with the older forms of slow-running pumps even if driven by steam, air or

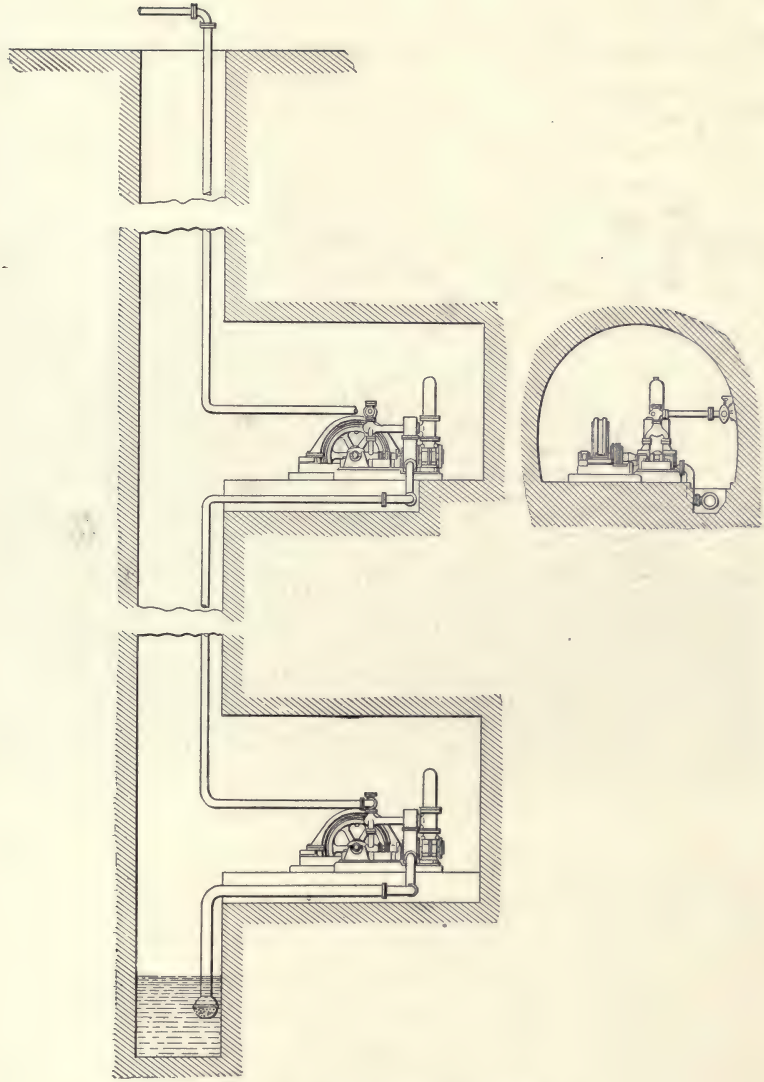


FIG. 496.—Express Pumps in Series.

pump rod. In this way considerable expense is saved in preparing a pump room.

The cost of the apparatus in this case has been very materially reduced. Two 1600-gallon independent high-speed pumps to work against 1500 feet head, and each driven by an 800 H.P. slow-speed induction motor cost $80,000 and they weighed with the motors 600,000 pounds. The same capacity of rod-driven pumps would cost $960,000 and the

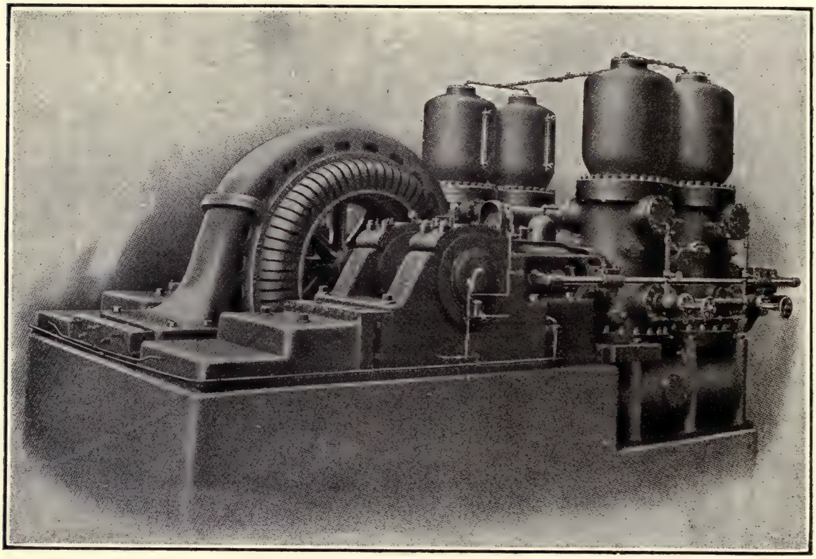


FIG. 497.—Knowles Express Pump.

moving parts alone would weigh over 5,000,000 pounds. The cost would be twelve times and the weight eight times as much as that of the high-speed electrically-driven pump.

The express pump of Riedler was explained on p. 163 and a section of it was shown in Fig. 154. After Riedler had shown that high speeds were possible with proper valve and proper air chambers, others took up the construction of these pumps and found that the mechanically-closed valves were not necessary and that self-closing valves could be used. In some cases the valves are of small diameter.

Fig. 497 is a Knowles express pump intended for mine service. The pump is a duplex double-acting pump, $6\frac{11}{16} \times 15$ inches which has a capacity of 1600 gallons per minute, against a head of 1550 feet. The motor is of 300 R.P.M. This speed is very high and the inertia of the water column is so great that there would be a continuous breaking of the column with its accompanying knocking if the air chambers and valves

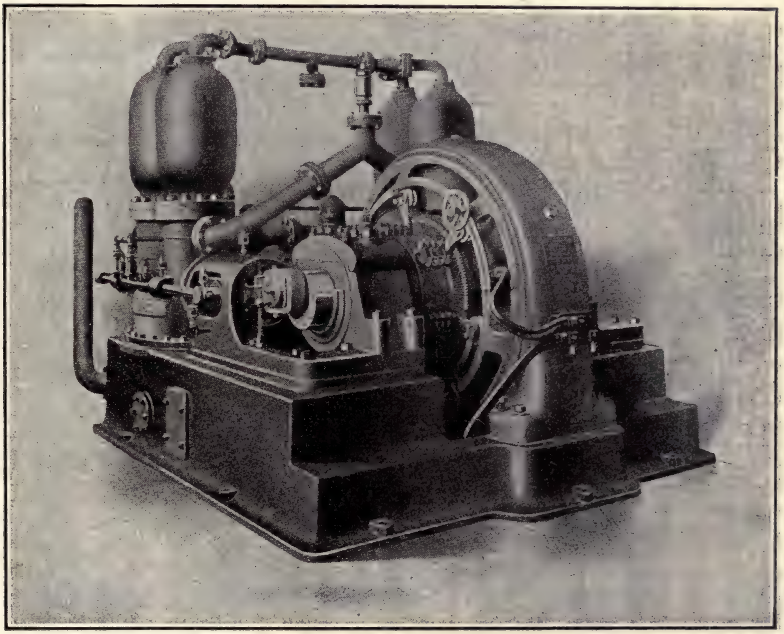


FIG. 498.—Knowles Express Pump.

were not properly designed. Poppet valves are the types used with this pump. They have the advantage of being almost balanced and of giving a large area of discharge.

The figure illustrates the heavy construction necessary on the outlet side, the air piping for charging the chambers, the outside rods to connect the front and back cross-heads, the pressure oiling system and the starting by-pass and priming valves. The general lines shown in the figure are excellent and the design is massive and so arranged as to

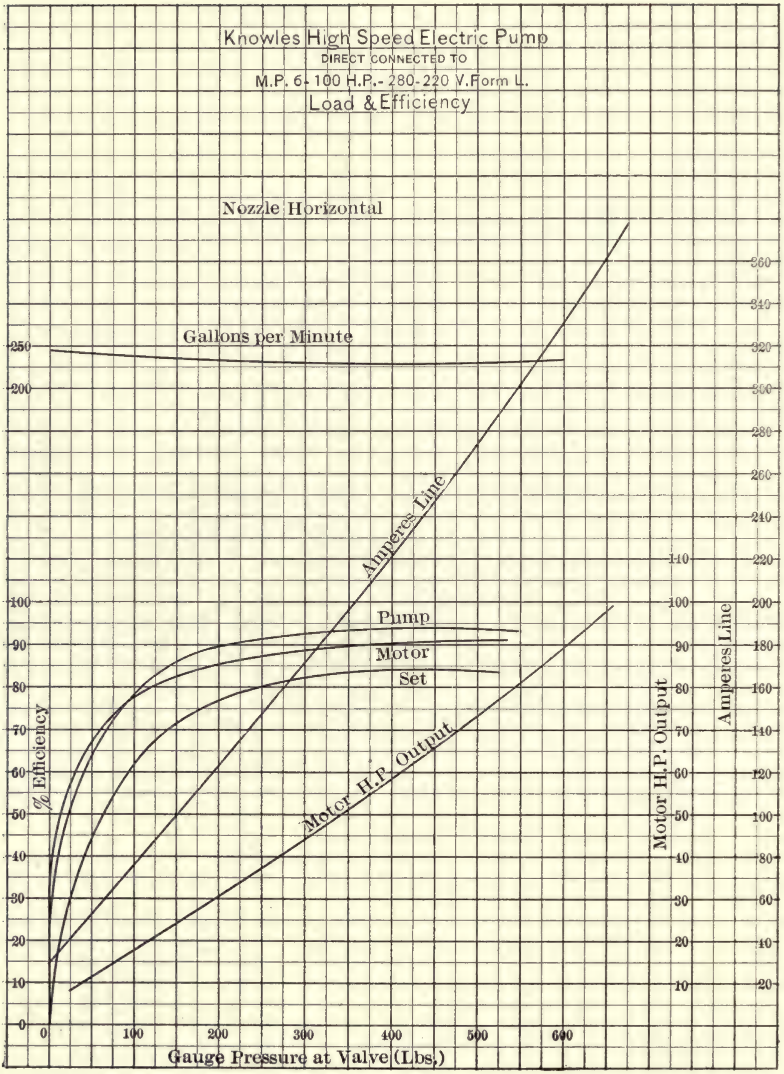


FIG. 499.—Test Curves of Express Pump.

care for the strains brought on during the operation of the pump.

The pump shown in Fig. 498 is a smaller unit, driven by a 100-H.P. direct-current motor. This pump is also built by the Knowles Steam Pump Works. It was intended to operate at 300 R.P.M., pumping 250 gallons per minute against a head of 1000 feet. The plungers were $3\frac{1}{2}$ inches in diameter and of $5\frac{1}{2}$-inch stroke. The machine is self-contained and the frames and bed-plate are constructed to make a rigid structure. The plungers, which are outside packed, are connected by outside rods.

A test of this pump gave the results which are shown by curves in Fig. 499. The result of 94 per cent efficency for the pump and 84 per cent for the combined unit, lifting water 1200 feet is remarkable. The electric line loss in operating such a pump may be rendered small by using a high voltage, when the installation becomes very efficient.

The types of direct-acting steam pumps used for mine work have been discussed in Chapter III and the forms illustrated by several figures. Fig. 500 illustrates another one of these pumps built by the Jeanesville Iron Works Co.

It is intended to lift 1200 gallons per minute against a head of 700 feet. The size of the valve pots, the control valves and gauges, the arms of the rotating steam valves, the method of valve operation, and the forms of the cylinders may all be clearly seen.

The valve pots are particularly large in this pump because of the large size of valves used by this company. Fig. 501 shows two forms of valves used by them for heads up to 750 feet. The annular valve is used when the head is not greater than 1000 feet. This figure illustrates the method of lining the valve pot with wood when the water contains acids which will attack the metal.

The design of mine pumps depends entirely on the type of pump. When the pump is of the fly-wheel type, the method is that used in Chapter V. The great length of discharge pipe would mean considerable inertia, but by the use of air chambers

the discharge in the pipe is not a fluctuating one and hence the friction in this pipe line is the only part requiring consideration in addition to the static head. The problem of the fly-wheel design is slightly different from that considered in Chapter V if an electric motor is used to drive the pump. In this case the tangential effort of the pump end pressure, combined with the friction of stuffing boxes, and forces from inertia and weight, is found. From this the excess and deficiency of tangential effort area is determined above and below a line

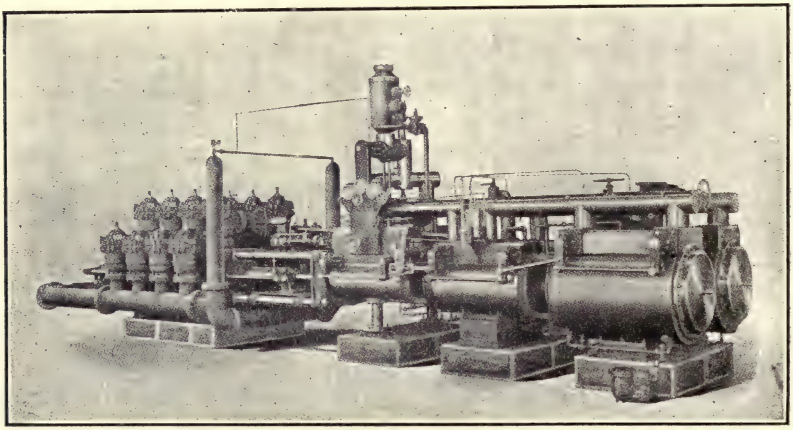


FIG. 500.—Jeanesville Mine Pump.

representing the delivered torque from the motor, which, of course, is constant.

The problem of a direct-acting steam pump without a fly wheel has to be handled in a different manner. The inertia of the moving parts is the important element in this design. The first problem to be considered will be one involving the type of mine pump shown in Fig. 494, but in which the fly wheel is omitted. In this case it is assumed that the indicator cards from the two steam cylinders are those shown at A, Fig. 502. These are combined as at B by taking the height a, b, to be

$$(a_1b_1)=[(p_{1h}-p_{2_h})A_h+(p_{1l}-p_{2l})A_l]\div A_l.$$

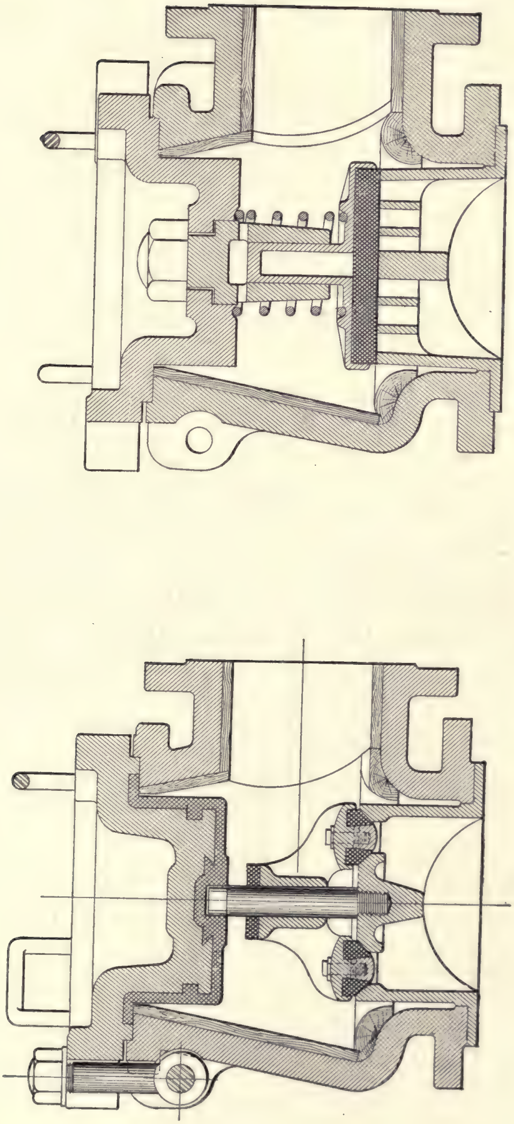


FIG. 501.—Jeanesville Valve Pots.

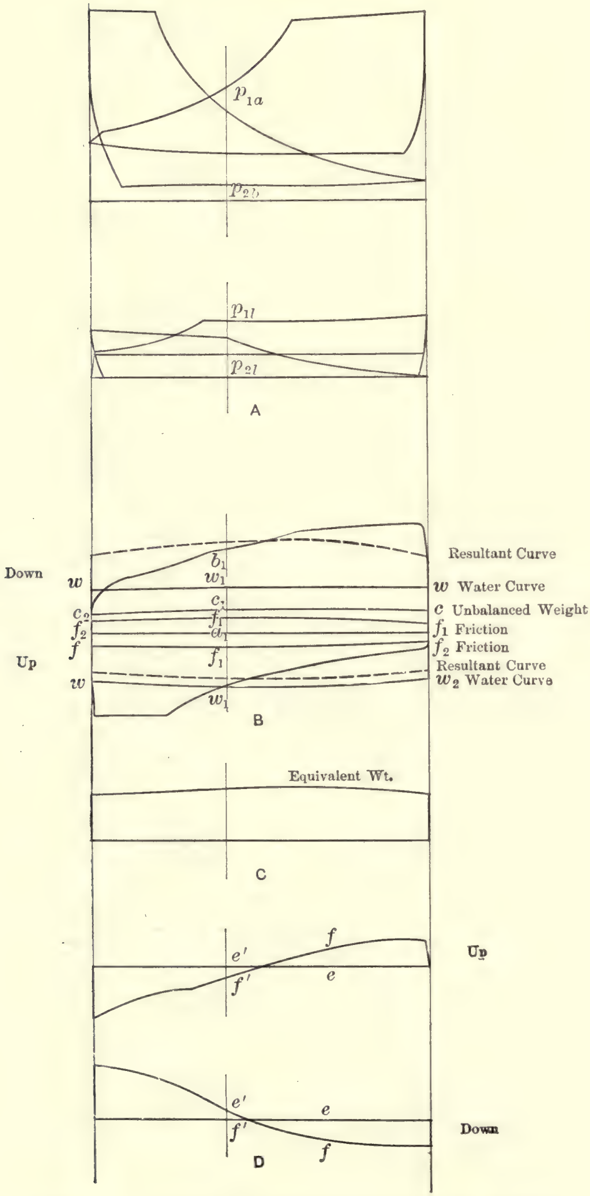


FIG. 502.—Forces in Direct-Acting Rod Mine Pump.

This is the force of the steam acting on the system. The unbalanced weight of the moving parts of the whole system per square inch of low pressure piston area is next found as

$$\frac{W}{A}=U.$$

The value of this force which reaches the piston rod will depend on the inclination of the bell-crank levers and by constructing the perpendiculars from the pivot, p_h and p_v to the line of pump-rod pull and piston-rod pull, respectively, the pull in the direction of the piston rod may be computed.

$$\frac{Up_h}{p_v}=U_r=a_1c_1.$$

This gives the curve c, c_1, c_2.

The weights of all the parts of the rods, balance weights, levers and pump pistons are now found. These must have an acceleration equal to the acceleration of the piston, multiplied by the ratios of the various lever arms up to the part considered. Thus the acceleration of the main pump rod is

$$\alpha=\frac{p_v}{p_h}\alpha_p.$$

If $\frac{a}{b}$ is the ratio of the arms of a balance lever the acceleration of this part is

$$\alpha_p\frac{p_v}{p_h}\frac{a}{b}.$$

If now each weight is examined and its acceleration found in terms of α_p, the force which must be exerted on the piston to accelerate all of these parts is

$$F=\Sigma\alpha_p\frac{p_v}{p_h}\frac{a}{b}\frac{W_x}{g}$$

$$=\alpha_p\sum\frac{p_v}{p_h}\frac{a}{b}\frac{W_x}{g}.$$

This means that if the sum of the terms $\sum \frac{p_v}{p_h}\frac{a}{b}W_x$ is found, this will give a weight which may be assumed to have the same motion as the piston in computing the force on the piston required to give these parts their motion. These ratios, $\frac{p_v}{p_h}$, $\frac{a}{b}$, etc., vary slightly, so that this sum will change as the piston moves. The value of this is laid off for different positions of the piston along the stroke, as in C, Fig. 502.

The pressure per square foot on the pistons from the water is equal to

$$wh = w\left[h_l + \frac{v_d^2}{2g}\left(1 + f\frac{l}{d} + m\right)\right],$$

if the air chambers are of the proper size to prevent the fluctuation of pressure, and the connections to the air chambers are so short that the inertia of the water in this connection can be neglected. If the connection is not short, then the weight of the water between the chamber and the pump must be added as a weight of one of the reciprocating parts. In general, however, the terms involving X and α of Eq. (21), Chapter V, are so small when compared with h_l that they may be neglected. The sum of the products $whA\frac{p_v}{p_h}\frac{a}{b}$ for the various pump pistons when divided by A_l will give the pressure per square inch which must be exerted per square inch of L.P. piston required to lift the water. This is shown by the curve w, w_1, w_2, etc., for the different piston positions.

If there is no air chamber on the pump, the weight of the whole column of discharging water must be added to the weight of the reciprocating parts in the determination of the weight to be accelerated.

The friction of the various balancing and supporting levers should next be found and with this the friction of the rods against their supports when not vertical as well as the friction of the stuffing boxes. These forces are overcome by a force

in the direction of the piston travel which may be found by the use of the moments of the various forces about different pivots. In this way the curve f_1, f_2, f_3 . . . is found, showing the force per square inch of low-pressure piston area which must be exerted on the piston in order to overcome the friction. These curves are combined, giving the resultant curve shown at D. The point 1 is found as

$$Ef_1 = a_1b_1 - a_1w_1 - a_1b - a_1c_1,$$
$$Ef' = a_1b_1 - a_1w_1 - a_1b + a_1c_1.$$

Since the term a_1c_1 is added in one case and subtracted in the other, it is seen that the work on the steam end is less on the one stroke than on the other unless the weights are balanced. If underbalanced, the work of the steam end is greater on the up stroke, while if overbalanced, the work of the steam end is greater on the down stroke.

The area of the curve $f'f_1$, which is the resultant of the pressures which act during the stroke, represents unbalanced work and hence the positive and negative parts must be equal, or the moving parts of the pump will not come to rest at the end of the stroke. In the actual engine of the direct-acting form it may be said that the stroke ends, the moving parts coming to rest, when the negative area equals the positive. On the return stroke the same must be true.

If now the net unbalanced pressure on each square inch of piston area at some point of the stroke be divided by the equivalent weight per square inch of piston area at that point and multiplied by g, the result will be the acceleration in the parts caused by the force, or

$$\alpha_p = g\frac{f}{W_t}.$$

If now a curve be plotted with the values of α_p for different points, the result will show how the acceleration varies.

This is shown in Fig. 503. Now

$$a = +\frac{dv}{dt} = \frac{dv}{ds} \times \frac{ds}{dt} = V\frac{dv}{ds}.$$

Hence

$$V\frac{dv}{ds}ds=ads,$$

and

$$\frac{dv}{ds}ds=dv,$$

$$\int V\,dv=\int a\,ds=\frac{V^2}{2}.$$

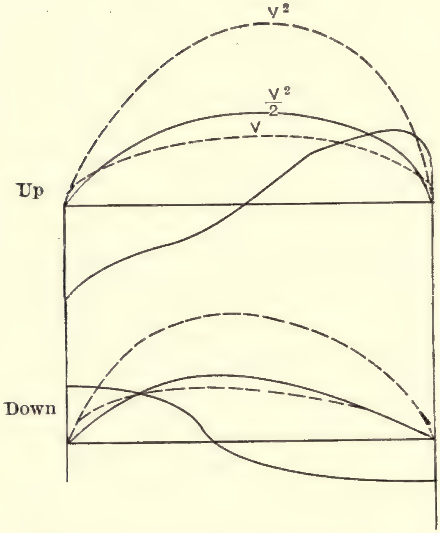


FIG. 503.—Acceleration and Velocity Curves.

The area of the curve just drawn from the end to any point is $\frac{V^2}{2}$. If then the integral curve be drawn in Fig. 503 the ordinates of this curve will be $\frac{V^2}{2}$ and from these V may be found, giving the dotted curve. From this curve the velocity time curve may be constructed as shown in Fig. 193 and from it, the time taken to complete one stroke. The curve is constructed for the two strokes. The time found will determine the number of strokes per minute and consequently the capacity of the pump.

If it is desired to cut down the speed of the pump more, a balanced mass could be added to the system, causing the acceleration curve to be lower, although the force diagram, Fig. 502, would not change. This may also be accomplished by decreasing the maximum steam pressure and carrying the cut off later. The total area of the indicator card cannot be changed, as that must be equal to the work done on the water and in overcoming the friction.

If it is desired to increase the speed of the pump, the balanced mass must be made smaller if possible and if this cannot be done the steam pressure is increased and the cut off is made earlier. The first of these changes the acceleration curve, increasing its height without any change in the curve of unbalanced force, while the second method increases the height of the acceleration curve by increasing the height of the curve of unbalanced force, since there is no change in the weight of the moving parts in this case. In the latter case also, it must be remembered that the area of the indicator cards cannot be changed.

The mass in some cases may be so great that even with high steam pressure the speed must be slow and the capacity small. In such cases balance weights may be omitted; then it becomes necessary to cut down the amount of steam used on the down stroke. A better method is that shown in Fig. 495, in which two rods and pump sets are so used that as one is descending the other is ascending. In this case the effect is a balanced one without the inertia effect of balance weights, for each side of the apparatus is designed to do pumping, and the mass is that of an unbalanced pump. Such an arrangement will give the fastest moving pump of this type.

In using these pumps it is well to note that should the steam pressure be increased without the reduction of the cut off, the positive area of unbalanced force would be greater than the negative area, and consequently the piston would strike the head with considerable force. For this reason a large cushion space should be used or springs should be applied to take up the shock. In case the pressure is reduced the

piston will come to rest as soon as the negative area is equal to the positive area. In some cases the steam may be admitted to the opposite end, at which time the full boiler pressure will act to stop the motion and reverse the pump. This would mean a large increase in the negative residual pressure, which would mean a rapid negative acceleration, bringing the parts to rest.

The use of direct-acting pumps with steam cylinders at the pump involves the same principles of design as the pumps just considered. In this case, however, the total mass reciprocated is quite small and for that reason the residual steam

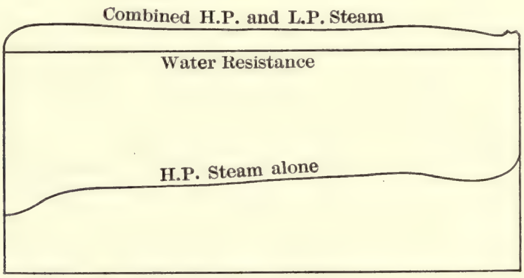


FIG. 504.—Compound Direct-acting Pump Diagram.

pressure cannot be as large as used with the other type of pump, since the acceleration

$$\alpha_p=\frac{Fg}{W_t},$$

would be very great. This would cause pounding and there would be some danger of wrecking the pump if the valve gear did not work properly. For this reason the method is to carry the pressure practically the full length of the stroke, getting the advantages of expansion in the use of several cylinders in which the cut off is at or near the end of the stroke. For such the combined card is shown in Fig. 504. The steam pressure is slightly above the resistance for the major part of the stroke and the piston is brought to rest by the cushion steam or the valve reversal.

The principles of Chapter V may be applied in this case for the resistance of the various parts, the terms which depend on the acceleration being so small as to be neglected in comparison with the height through which the water is lifted. Should this be such that these terms are not small, they would have to be neglected in the first approximation, when the motion is found; then computed for the motion thus determined, and a second approximation made in which they are used. If these terms are changed much by the new curve of time and velocity a third approximation should be made.

In this manner the probable capacity of the pump may be found. Of course in this type of pump with a proper cushion chamber the speed may be increased by increasing the steam pressure. After the speed reaches its limit, pounding begins, as the cushion cannot care for the energy stored up in the moving parts.

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Am. Mch.	American Machinist.
Engng.	London Engineering.
Eng. Mag.	Engineering Magazine.
Eng. News	Engineering News.
Eng. Rec.	Engineering Record.
Engr.	The Engineer (American).
A.S.C.E.	Transactions of the American Society of Civil Engineers.
A.S.M.E.	Transactions of the American Society of Mechanical Engineers.
J.A.S.M.E.	Journal of the American Society of Mechanical Engineers.
N.E.W.W.	Journal of the New England Water Works Association.
A.W.W.A.	Transactions of the American Water Works Association.
Z. f. d. g. Turb.	Zeitschrift für das gesampte Turbinenwesen.
Z. d. V. d. Ing.	Zeitschrift des Vereines deutscher Ingenieure.
Elec. World	Electrical World.
Power	Power and the Engineer.
A.E.S.	Journal of Associated Engineering Societies.
Cassiers	Cassiers Magazine.

These are arranged under the following heads:

Texts.
Air Pump and Screw Pump Tests.
Centrifugal Pumps, Theory and Design.
Centrifugal Pumps.
Centrifugal Pump Stations.
Centrifugal Pump Tests.
Costs.
Fire Pumps and Stations.
Historical.
Injectors and Direct Steam Pumps.
Irrigation, Drainage and Sewage Pumps and Stations.
Mine Pumps.
Reciprocating Pumps, Tests and Results.
Reciprocating Pumps, Design and Theory.
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Simplex Pumps.
Special Pumps.
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